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INDICATOR DIAGRAMS

AND

ENGINE AND BOILER TESTING.

BY

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ERRATA.

Page 41.—Ninth line from top, read "fig. 40" instead of "fig. 39."

" 46.—In formula, read "R^2" instead of "R."

" 46.—In twentieth line from bottom, read "line A C *to* any point *in* B C" instead of as written.

" 46.—In fourth line from bottom, read "fig. 42" instead of "fig. 41."

" 46.—In third line from bottom, read "fig. 43" instead of "fig. 42."

PREFACE.

THE purpose of the first portion of this book—*i.e.*, the part relating to "The Indicator and Its Diagrams"—is to explain the construction of the different kinds of indicator used, and show their advantages and disadvantages, also to explain clearly the general principles on which the interpretation of indicator diagrams is based.

A large number of defective diagrams are illustrated and described, but the writer hopes that readers will not endeavour to commit to memory the special forms resulting from the various defects, but rather use them as instances of the modes of reasoning applicable to indicator diagrams generally. It might be well to mention here that, except where stated otherwise, the illustrations are exact reproductions to a smaller scale, of diagrams actually taken.

As regards the second portion of the book, "The Testing of Engines and Boilers," the writer believes that it will supply a want on the part of those who have not already had experience in such work, as the information previously obtainable is very scattered, and in some cases difficult of access.

The very complete Table of Piston Constants, compiled by Mr. W. H. Fowler, will doubtless be found very useful.

Should any errors be observed at any parts of the book, the author will esteem it a favour if those noting them will apprise him.

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THE INDICATOR AND ITS DIAGRAMS.

CHAPTER I.

Development of the Indicator.

The steam-engine indicator, now so indispensable in connection with all steam-engine work, was invented by James Watt, and it is evident that he fully realised the importance and usefulness of the instrument. It enables us, amongst other things, to ascertain the pressure of steam on the engine piston at all points of the stroke. A steam gauge would not perform this service satisfactorily, as, although the pressures themselves might possibly be correctly recorded, it would be necessary to note exactly the position of the piston for each pressure, which would be quite impossible. This difficulty is got over in Watt's and other indicators by what we might term the pressure-gauge portion of the indicator registering its pressure on a suitable card fixed on a frame of some kind, and caused to move coincidently with the piston. Thus we get a simultaneous reading, showing the pressure in the cylinder and the position of the piston.

The instrument invented by Watt consisted of a small cylinder, about 1 in. diameter, having a piston carefully fitted in it. The piston rod connected to this worked through a guide, so that the motion of the piston might be quite free. At the lower end of the cylinder was a tap, which could be connected to the engine cylinder. When this was done, communication could be made between the indicator and engine cylinders by simply opening the tap. On the upper side of the cylinder was a steel spiral spring, the upper end of this being fixed, and a pencil was attached to the top of the piston rod, this pencil being arranged so as to mark on a paper fastened to a board, caused to slide backwards and forwards by a string connected to a suitable part of one of the parallel-motion radius rods, the motion of the sliding board corresponding, therefore, to the motion of the piston. A string and weight or a spring were also attached to the board, to pull it backwards and keep the driving cord always in tension (see fig. 1).

Suppose that an indicator of this description be attached to the top end of a beam-engine cylinder, and that a driving

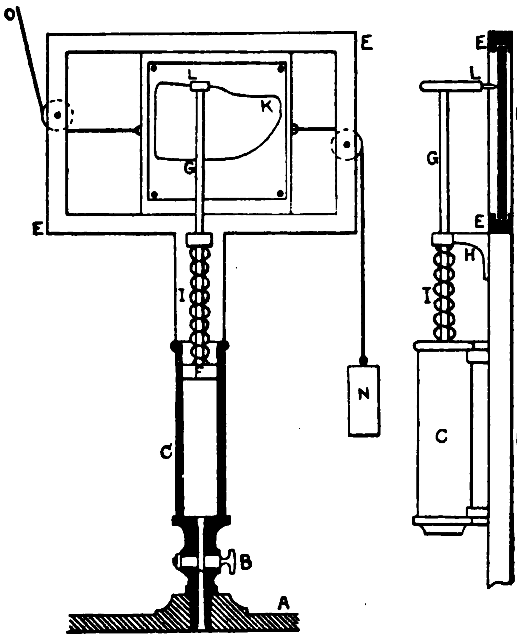


FIG. 1.—Watt's Indicator.

A is portion of the engine cylinder.
B is the indicator tap.
C is the indicator barrel or cylinder.
D is the supporting board for whole arrangement.
E is the frame in which the diagram board slides.
F is the piston of the indicator.
G is the indicator piston rod.
H is the piston-rod guide.
I is the spring.
K is the board to which the paper is pinned.
L is the pencil attachment to the piston rod.
N is the weight for moving the board K to the right, as permitted by the driving cord.
O is the cord for driving the board, so that its motion may be exactly similar to the motion of the engine piston.

cord be connected, as described previously. On opening the tap B the indicator piston will be at once subjected to

the same pressure as the engine piston. If the pressure were higher than atmospheric, the indicator piston would be forced upwards until the surplus pressure became balanced by the compression of the spring above. If, on the other hand, the pressure in the cylinder were below the pressure of the atmosphere, the indicator piston would descend until the tension of spring which such descent would cause balanced the vacuum in the cylinder—*i.e.*, the difference between the cylinder and atmospheric pressures.

From this we see that the pressure in the cylinder will, at all times, be registered by the pencil attached to the indicator piston rod, and consequently, if the strength of the spring is known, it becomes easy to measure this pressure. It is usual to so proportion these springs that each pound of difference in the pressures causes the pencil to rise or fall some readily-measured dimension, such as $\frac{1}{8}$ in., $\frac{1}{16}$ in., $\frac{1}{10}$ in., $\frac{1}{20}$ in., &c.

The indicator pencil is placed so as to mark a sheet of paper fixed to the sliding board K, whose motion is precisely similar to the motion of the piston—that is to say, the board is at the end of its travel when the piston is at the end of its, and similarly for every other position. In consequence of this, markings of the pencil represent not only the pressure on the piston, but the pressure on the piston at each position of the stroke, whether travelling upwards or downwards. The diagram thus taken becomes a valuable record of what was taking place in the cylinder of the engine.

The principles involved in Watt's indicator are also those of the most modern indicators, the only difference being in details rendered necessary by the altered conditions under which the engines themselves have now to work.

One of the principal defects felt in connection with Watt's indicator was its cumbersomeness, it being heavy and inconvenient to handle; also the sliding board with its weight did not form a desirable arrangement apart from inconvenience, as will be seen further on when discussing the modern indicators.

The first important improvement on Watt's indicator was made by McNaught, who substituted the sliding frame and board by a revolving cylinder or drum, the diagram paper being wrapped round the cylinder and held in position by a pair of clips. The barrel was pulled round by a cord, corresponding to O in Watt's arrangement, while to pull it back, and keep the string in tension, a volute steel spring

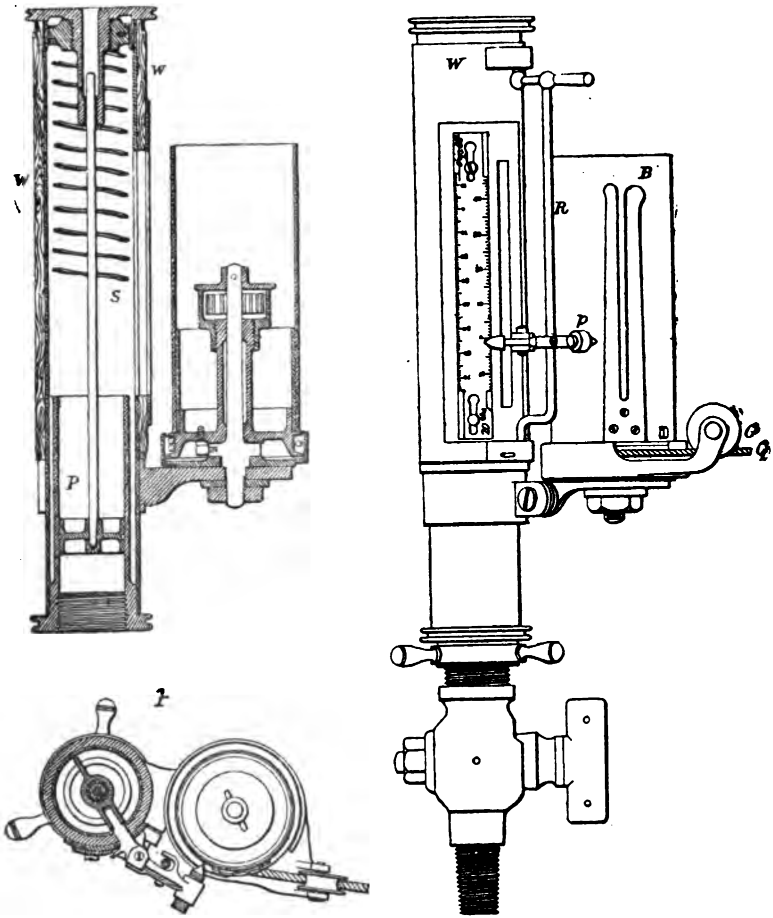


FIG. 2.—McNaught's Indicator.

P is the indicator piston.
S is the spring attached to the piston.
B is the barrel for carrying the paper.
***p* is the pencil for marking the paper.**
R is a rod used for placing the pencil against the paper.
C is the driving cord.
G is the guide pulley for the driving cord.
W is the casing enclosing the spring.

was placed inside the drum. McNaught also enclosed the piston spring by an extension of the cylinder, and he attached his pencil to the lower end of the spring. The general arrangement will be easily understood from fig. 2.

It will be readily seen that although the diagram paper now moved to and fro round the axis of the paper drum, instead of along a slide, the diagram traced on it would still represent the pressures at each point of the piston's stroke as truly as before ; at the same time the indicator was much more portable and handy.

McNaught's indicator was the first one which came into general adoption, and it was largely used on the slow-running engines of its day, in connection with which it proved very satisfactory ; and it may even be occasionally met with at the present day. This indicator, however, proved unsuitable for taking diagrams from engines running at anything like a high speed, owing to undue vibration of the pencil ; the cause of which is easily understood when we consider what takes place in the cylinder of the engine.

Suppose that the engine piston is moving towards the front end of the cylinder, then it will be forcing the exhaust steam out through the front port, and this will go on until the piston is within, say, several inches of the end of its range of movement. The exhaust port will then be closed by the valve, and compression will take place until the piston is just about the end of its stroke, when the steam port will be opened, and high-pressure steam will rush into the cylinder, thus causing an almost instantaneous increase of pressure.

The result of this sudden increase of pressure, as regards the indicator, will be to cause an equally sudden upward movement of the piston, piston rod, pencil, &c. ; also to some extent of the spring itself, which in McNaught's indicator was of considerable length. The consequence of this will, as will readily be seen, be that these moving parts will have acquired considerable momentum before the spring is compressed to a pressure corresponding to the steam pressure in the cylinder, and in consequence of this they will compress the spring still further, and show a pressure on the diagram much greater than actually occurs in the cylinder. Again, after they have been brought to rest at the top, the surplus pressure of the spring will cause them to descend rapidly, and the pencil will this time be carried below the true pressure by the momentum of the moving parts. It is quite possible that these vibrations of the pencil

may go on with gradually decreasing range until the piston has reached the other end of the stroke. Sometimes, if the terminal pressure is high, the rapid fall of pressure taking place on the opening of the exhaust port will again cause vibration of the indicator pencil.

Illustrations of these defects are shown at fig. 3.

Fig. 3 is a diagram taken by a McNaught indicator from an engine running at 80 revolutions per minute, and shows clearly the oscillation of the pencil owing to the momentum of the moving parts.

A brief consideration will show that such vibration is greatly influenced by the following: (1) Speed at which the engine runs, as the higher the speed the more suddenly will the pressure come on the piston of the indicator, and the greater will be the portion of the stroke during which the vibrations occur. (2) The weight of the moving parts of

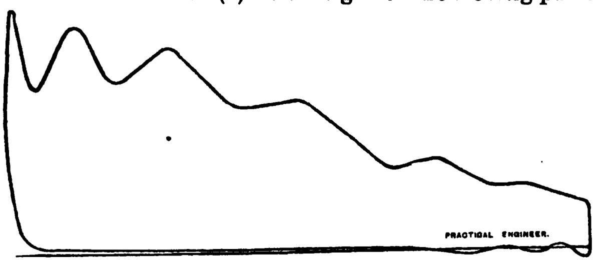


FIG. 3.—Diagram showing Effect of Oscillation of Spring.

the indicator, as the heavier the parts the greater will be their momentum, and consequently the greater the vibration—that is, of course, for pistons of equal size. (3) The strength of the spring in the indicator. If a weak spring were used, the indicator piston would have to move through a greater space to register the required pressure than if a stronger spring were used, and consequently the vibration would be greater. Hence, if such vibration shows itself on any diagrams, a fresh set should, if possible, be taken with a stronger spring in the indicator. By using a strong enough spring the vibration may be practically quite eliminated, even with McNaught indicator; but it is quite possible that, owing to the small movement of the pencil—which, with this indicator, would be the result of using such a strong spring—the diagrams would be so small as to become of little value. In illustration of this point the diagrams

shown by fig. 4 are introduced. These have been drawn on the assumption that a spring four times the strength of that actuallly used was put in the indicator. Under these circumstances we might reasonably expect no more vibration than is shown, but it is at the cost of obtaining a diagram so small that the various points in the steam distribution are very indistinctly shown, and so small that calculations of mean pressure would be very liable to great inaccuracy. (4) The friction of the indicator also greatly influences vibration of the pencil. If the friction is increased, the vibrations will be lessened; in consequence of this it is not uncommon to find people, when indicating, pressing the pencil hard on the paper, so that the increased friction may lessen vibration. This course, however, should certainly not be adopted, as the increase in the friction causes the diagrams to be materially incorrect. (5) Considerable compression lessens the risk of vibration by diminishing the rise of pressure at admission. If the exhaust steam were compressed just up to the initial pressure, no rise of pressure would occur when the steam port opened; hence there

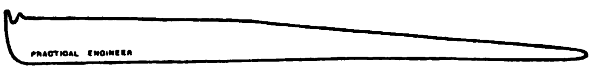


FIG. 4.—Diagram showing Limited Range due to Excessive Strength of Spring.

would be no risk of oscillation. From this it will be seen that an indicator might give satisfactory diagrams from an engine running at a certain speed with good compression, but not even passable diagrams if the compression were very slight, although the speed remained unchanged.

The ordinary spring letter balances found in most offices may be used to illustrate the foregoing remarks respecting vibration in indicators.

Take such a balance, preferably one having a weak spring—say, which allows a movement of about an inch per ounce, and which will register at least two ounces. Place a weight of an ounce suddenly in the pan of this balance, and it will be seen that the index pointer is a considerable time before it comes to rest, and that it vibrates on each side of the true weight, the oscillations becoming gradually less and less until the pointer takes up its proper position.

This represents generally what occurs in the steam-engine indicator, only that all the vibrations are marked on the diagram paper, and the steam pressure takes the place of the weight.

Take now another balance, having a stronger spring, and repeat the experiment, and it will be seen that the oscillations are much less, and do not last as long as in the former experiment; hence the advantage of using strong springs.

The next important improvement in indicators was introduced by Mr. Charles B. Richards, of Hartford, Connecticut, in 1862, and consisted essentially of an arrangement for multiplying the motion of the piston, so that the pencil might mark a large diagram, although a strong spring was being used in the indicator; and in this respect the Richards indicator was the pioneer of all the modern indicators.

The multiplying motion adopted by Richards was an adaptation of Watt's well-known parallel motion (see fig. 5). By this means the pencil was caused to move in a straight line, parallel to the axis of the indicator cylinder, and four times as fast as the piston—that is to say, a $\frac{1}{8}$ in. movement of the piston would result in a movement of the pencil of $\frac{1}{2}$ in.—*i.e.*, four-eighths. The arrangement for carrying the paper remained as in McNaught's indicator.

From this explanation the reader will see that the disadvantages attending the use of strong springs were overcome, and consequently the tendency to vibration, when taking diagrams of ordinary size, was greatly diminished. Owing to this improvement, indicator diagrams much more truly represented the action of the steam in the cylinder, with the result that the Richards indicator has become an instrument of almost every-day use in the hands of a very large number of engineers, and the great perfection to which steam engines have now arrived may be attributed greatly to this fact.

Fig. 5 shows a sectional elevation and a plan of a Richards indicator, from which the general construction will be noted. The paper drum is identical with that used in McNaught's indicator, as shown at fig. 2.

The two small holes shown in the cylinder cover, in the plan of the indicator, are to allow of the free escape of such steam as may leak past the piston. Providing the holes are sufficient to allow any such steam to escape freely, leakage past the indicator piston is not objectionable, unless it becomes so great as to render it difficult to handle the instrument. On the other hand, slight leakage is rather advantageous, as it assists in lubrication and lessens the friction.

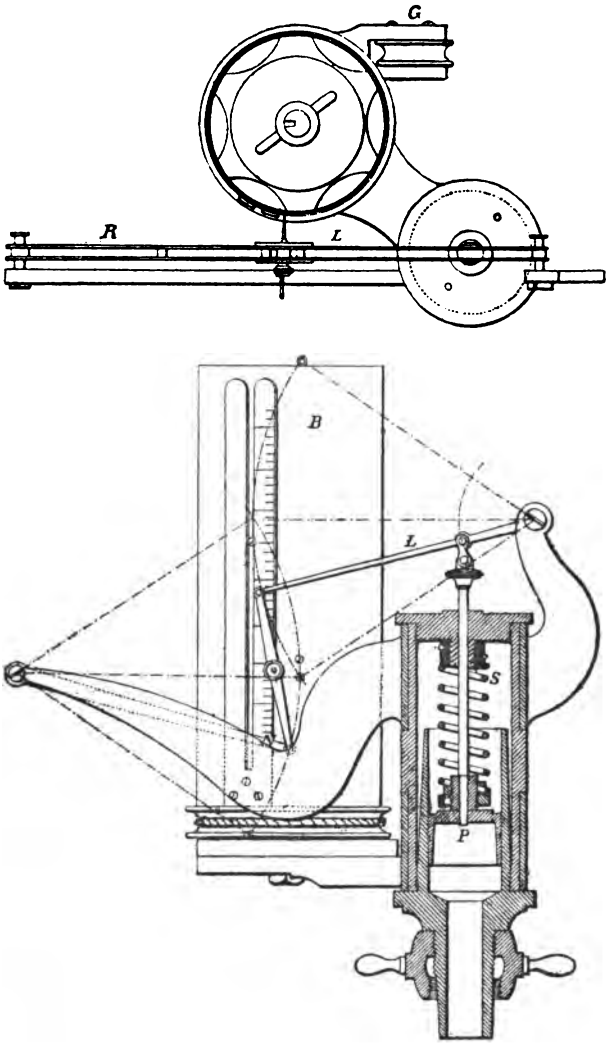


FIG. 5.—Richards Indicator.

P is the piston,
S is the spring.
C is the cylinder.
B is the paper barrel.
L is the parallel motion lever.
R is the parallel motion radius rod.
G is the guide pulley for the driving cord.

The lower end of the spring S screws on one part of the piston, whilst the upper end screws on an extension of the cylinder cover.

To remove the spring for any purpose, proceed as follows: Unscrew the small milled nut on the top of the piston rod, then unscrew the cylinder cover, and withdraw the cover, spring, and piston together; after this unscrew the spring from the cover, and then from the piston. When replacing the spring, proceed in exactly the reverse manner, always taking care that the spring is properly screwed home, both on the piston and on the cylinder cover, but being careful not to strain the spring. Before re-inserting the piston it should always be carefully cleaned and oiled with suitable oil.

The Richards indicator has been found quite reliable, when using a strong spring, on engines making up to 130 or 140 revolutions per minute, and higher under favourable conditions; but a large number of engines at the present day run at a much faster speed than this, and in connection with them this form of indicator has not proved altogether satisfactory, owing, again, to vibration of the pencil. Hence the principal object aimed at in the more modern indicators has been the lessening of vibration when indicating at very high speeds.

The Thompson, McInnes, Crosby, and Tabor indicators are instances of such instruments. In each of these the principle introduced by Richards—viz., that of using a strong spring, and compensating for this by levers arranged so as to multiply the piston's movement—has been strictly adhered to, the main changes being in the design of the multiplying arrangement or parallel motion, and of the details of the piston, piston rod, &c., so as to reduce the weight of the reciprocating parts.

The Thompson indicator is shown at fig. 6, a glance at which will show that the parallel motion is much lighter than Richards'. The other reciprocating parts are lighter, and generally the instrument is better adapted to give reliable diagrams at a high speed than the Richards indicator.

The multiplying gear gives the pencil a movement of four times that of the piston, but the stroke of the piston is about $\frac{1}{8}$ in. less than Richards', being $\frac{3}{4}$ in.; consequently, the maximum height of the diagram is 3 in. instead of $3\frac{1}{2}$ in. From this it will be seen that the improvements as regards the suitability of the instrument for high speeds consist mainly in the decreased weight of the parts, and

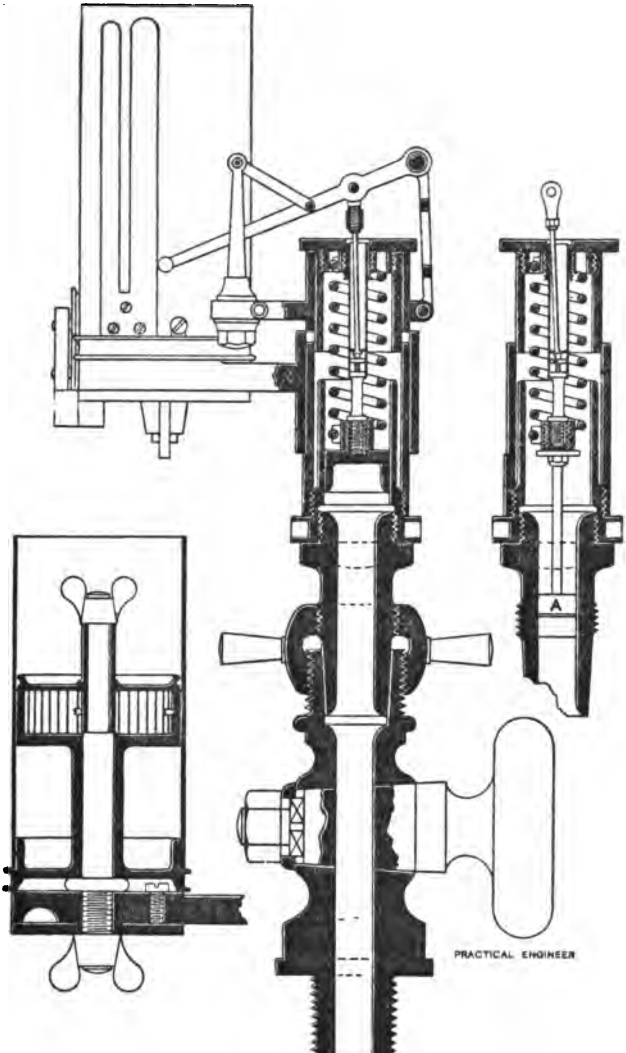


FIG. 6.—Thompson Indicator.

the fact that the design permits of those levers which have a large movement being made light. In the Thompson indicator the tension of the spring in the paper drum is easily adjusted by a neat arrangement. The advantage of having such adjustment will be explained further on.

Messrs. Schäffer and Budenberg have recently introduced a modification of the Thompson indicator, with a view of rendering it specially suitable for very high pressures, without in any way affecting its suitability for ordinary work. This arrangement consists of an extended piston rod, having a piston at its end of a diameter one-half that of the ordinary indicator. (See A, fig. 6.) This piston fits in a cylinder bored concentric with the larger cylinder, the smaller cylinder being in reality formed in the lower part of the indicator leading to the tap. The object of the invention has been to do away with the necessity for very strong springs, which are difficult to make correct, and are more liable to change than weaker ones. By means of this arrangement diagrams have been satisfactorily taken from hydraulic pumps working at pressures of 3,000 lb. per square inch. The makers of this indicator have also patented a form of indicator arranged to show the net effective pressure on the piston throughout the stroke (fig. 7). To ascertain this pressure accurately by means of ordinary indicators, it is necessary, especially if the load is of an irregular character, to use two indicators, one fixed at each end of the cylinder, and then take diagrams from the opposite ends simultaneously, afterwards measuring the accelerating pressures from one diagram and the retarding pressures from the other, and taking the difference as the net effective pressure.

The instrument in question (fig. 7) practically consists of a combination of two indicators, acting on one pencil simultaneously in such a manner as to record on the indicator paper the resultant or effective pressure on the engine piston. The steam from one end acts on the under side of one of the pistons, while the steam from the other end acts on the upper side of the other piston, the difference between the two pressures being the pressure required to be balanced by the indicator spring. The piston rod is so arranged that the spring is put in compression by downward as well as upward movements of the rod.

When taking diagrams to show the effective pressures, the indicator tap at each end of the cylinder must be opened. By only opening one tap at a time, an ordinary

indicator diagram would be obtained. The parallel motion and the paper drum are similar to those used in the ordinary Thompson indicator.

The Crosby indicator has latterly come very largely into use for indicating at very high speeds. This instrument is shown at fig. 8. The parallel motion is somewhat similar to

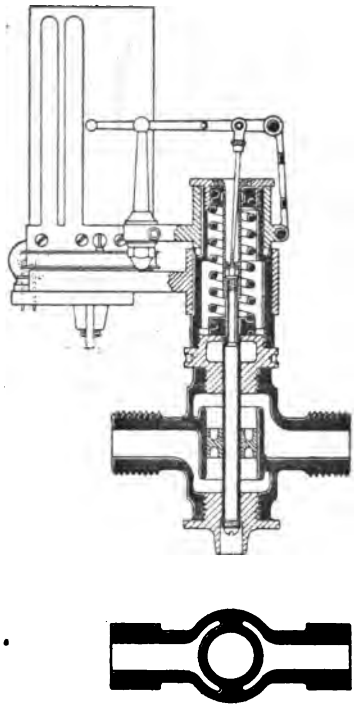


Fig. 7.—Thompson Double Indicator.

that on the Thompson indicator, except that the small guide link is attached to the vertical link, instead of the main lever. This lightens the motion work slightly, but it will be noted that the accuracy of the parallel motion is dependent on the true movement of the piston rod, so that, if through wear or defective workmanship the centre line of

the piston rod were not truly coincident with the centre line of the cylinder, the multiplying or parallel motion would not be strictly accurate.

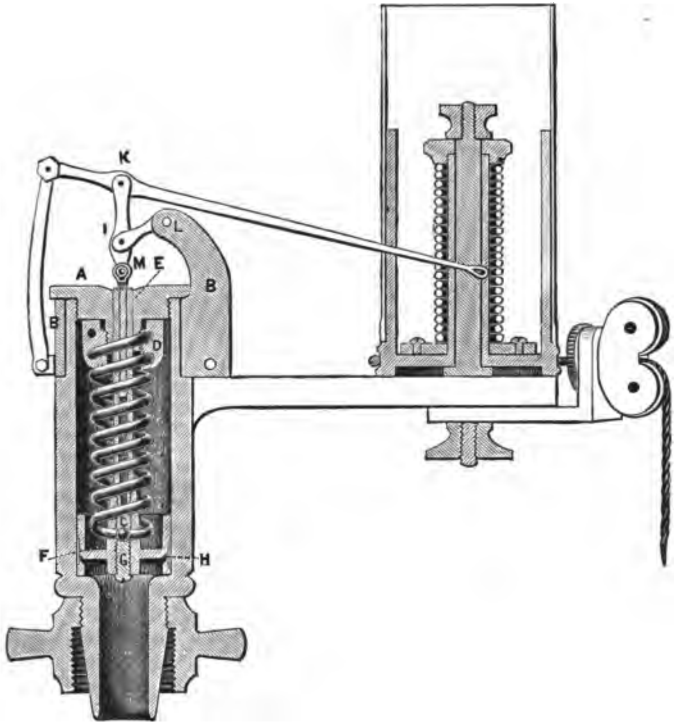


Fig. 8.—Crosby Indicator.

In the illustration of the Crosby indicator, fig. 8,

A is the cylinder cover or cap.
B is the sleeve which carries the motion work.
C is the ball connection of the spring to the piston.
D is the connection of the spring to the cover A.
E is the screwed rod connection, the multiplying gear to the piston.
F is the piston.
H is the groove cut in the piston to lessen friction and leakage.
I is the vertical connecting link.
K is the pencil lever.
L is the radius link.

The wear, however, at the piston-rod guide is so very slight under ordinary working conditions that it may be safely disregarded, whilst the second objection may be got over by careful workmanship.

This indicator is so arranged that the movement of the piston is only about $\frac{3}{8}$ in., the movement of the pencil being six times this amount; hence the height of the diagram is about $2\frac{1}{4}$ in.

From the explanation previously given the reader will understand why this reduced motion of the piston is advantageous in reference to high-speed indications, and that when it is combined with light moving parts the indicator is rendered specially suitable for such work.

An objection sometimes raised against the Crosby indicator is the smallness of the diagrams taken with it. This objection certainly has weight as regards engines running at only moderate speeds, where large diagrams may be satisfactorily obtained, but an accurate small diagram is decidedly better than a larger one not quite accurate; hence for engines working at very high speeds it is wise to sacrifice largeness of diagrams for a more accurate outline.

Another objection to indicators of the high-speed class is that their extreme lightness of construction necessitates them being carefully handled at all times, and thus renders them less suitable for every-day work. In consequence of this, many engineers prefer to use an indicator of the Richards type for indicating engines running at moderate speeds, and turn to one of the high-speed indicators for taking diagrams from engines running at high velocities.

The screwed rod E forms a very neat arrangement by which the position of the atmospheric line on the indicator paper can be readily adjusted at will, as it is only necessary to unscrew the cap A from the cylinder body, and then turn it to the right or left, according as it is desired to lower or raise the pencil.

In the Tabor indicator, shown at fig. 9, a quite novel form of parallel motion has been adopted, the radius links of the Thompson and Crosby indicators being substituted by a plate having a curved slot, in which a roller attached to the pencil lever moves. The slot is so arranged that the pencil moves in a line parallel to the axis of the cylinder, and at the same time its speed of movement bears one definite proportion to the speed of the piston throughout its stroke, namely, 5 to 1.

The parallel motion of this indicator is very light, which

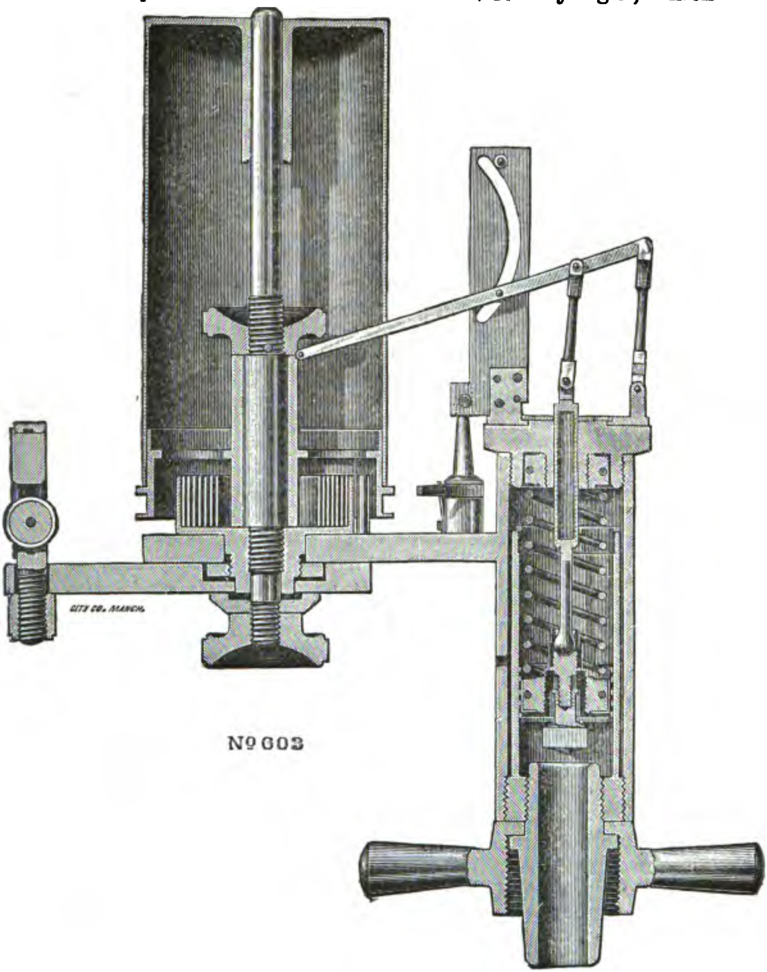


Fig. 9.—Tabor Indicator.

renders it suitable for high speeds, whilst at the same time

it has the advantage over the Crosby indicator of producing larger diagrams, these being practically the same size as are taken with an ordinary Richards indicator.

The piston springs in the Crosby, McInnes, Thompson, and Tabor indicators are of the double-coil type. By this means the pressure of the spring acts equally on two points on opposite sides of the piston, instead of being received on one side only, as is the case with springs of the single-coil type, and in consequence of this the frictional resistances are somewhat reduced.

A further precaution for the prevention of any cross or side pressures is taken in the Crosby indicator by connecting

FIG. 10.—Crosby Indicator Spring.

the bottom end of the spring to the indicator by means of a ball bearing, and this at the same time forms a lighter connection than the usual brass head. (See fig. 10.)

Double-coil springs have a further slight advantage over single ones for high pressures, as they are said to be easier to construct accurately, on account of being constructed of lighter wire than single-coil springs of the same strength.

The McInnes indicator has a parallel motion somewhat similar to that used on the Thompson indicator, and is shown at fig. 11. The cylinder is designed so as to have a large opening at the bottom end to facilitate cleaning, also the drum spring is made adjustable, but the special feature of the indicator is that it is sheathed in vulcanite, which makes it much more comfortable to handle.

The maximum size of diagram is 4½ in. long by 2 in. high, and the makers state that diagrams this length can be taken from an engine running at so high a speed as 400 revolutions per minute; with a shorter diagram and a stronger spring in the cylinder, engines running at still higher speeds may, it is stated, be satisfactorily indicated.

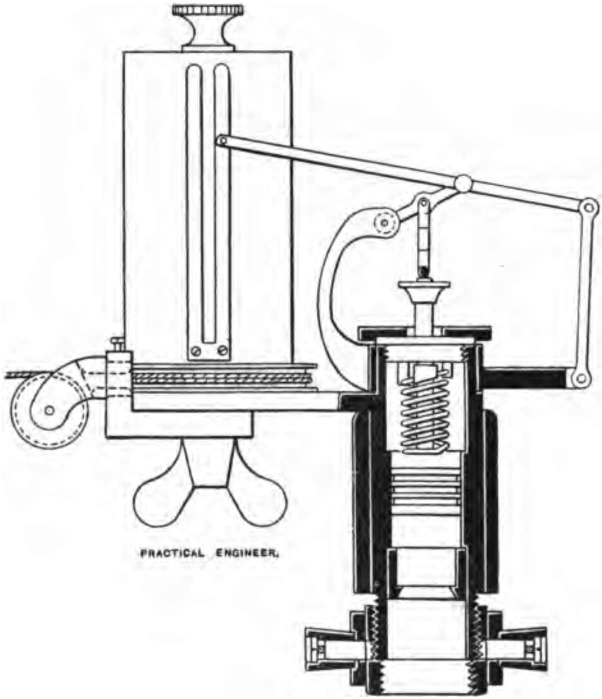


FIG 11.—McInnes Indicator.

For speeds above 200 revolutions per minute, they, however, recommend the use of springs, which will give diagrams about 1½ in. high.

There is still another point in connection with the indication of engines working at high speeds which requires attention, namely, the effect of such speeds on the movement of the drum carrying the indicator paper.

The drum is, in almost every case, actuated by a cord, which receives its motion from the crosshead through a reducing gear, so arranged that the movement of the point to which the cord is attached exactly corresponds, but to a reduced scale, with the movement of the engine piston. Now, we know that the velocity of the piston is a very variable quantity, as it comes to rest at each end of the stroke and reaches its maximum velocity about mid-stroke; hence it follows that the velocity of the paper drum is equally variable.

Suppose that we take, for example, a case in which the paper drum has been made very heavy, with a view to strength, whilst the drum spring is light; then at the commencement of a stroke the indicator cord will have the work of setting the barrel or drum in motion, and seeing

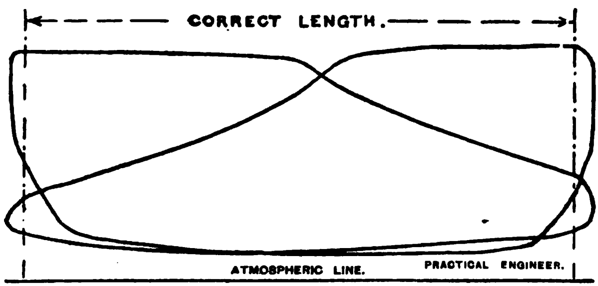


FIG. 12.—Diagram showing Distortion caused by Overrunning of Indicator Barrel.

that the drum is heavy, and that the resistance of the spring has also to be overcome, it is evident that a considerable stress will be put on the driving cord, and the stress on this cord will be greater than that actually required to overcome the tension of the spring alone so long as the velocity of the drum is being increased, but the extra pull will be a gradually diminishing quantity.

When the piston passes the centre of its stroke its velocity will commence to fall; consequently that of the drum should do likewise; but owing to this part being of considerable weight, its momentum will become important, and it is quite possible to conceive that the momentum will, at high speeds, be sufficient to overcome the resistance of the spring, and cause the drum to overrun the driving cord, thus giving misleading diagrams.

In illustration of this point, the diagrams shown by fig. 12 have been reproduced. These were taken from a horizontal non-condensing engine, running at 90 revolutions per minute, by means of an ordinary Richards indicator. The true length is shown on them. This was obtained by shutting off the steam supply, and carefully marking the length of the diagram when the engine was moving very slowly, with the result shown.

From the foregoing we see that the pull on the indicator cord is very variable ; consequently it is important the cord be as inelastic as possible, and owing to this it has become not uncommon to use flexible wire. If this is not convenient, a specially woven cord, which has been well stretched, should be used.

It would, however, be much better, if possible, to so proportion the drum spring that the tension on the driving cord may be very nearly uniform, and a little consideration will show that such a result may be approximately attained with proper precautions.

In the example taken we saw that just at the commencement of the stroke the pull on the cord was at its greatest, and that it was gradually decreasing, until when approaching the end of the stroke the pull might be entirely removed, owing to the momentum of the drum.

The entire removal of the pull might be obviated by making the drum extremely light and using a stronger spring ; but assuming that the tension of the spring is almost uniform throughout its movement, as would be the case if a long coil spring were used, the pull on the cord would still be very variable ; if, however, a short spring were used, the tension of this would rapidly increase as the drum moved round, and would therefore tend to counterbalance the variation of pull due to the inertia of the drum.

In the Crosby and McInnes indicators the usual long coil or volute spring is substituted by a spiral spring whose tension is readily adjustable, and in consequence of this the pull on the indicator cord can, with proper care, be kept nearly uniform.

A great source of difficulty in connection with indicators, and one which has given rise to much discussion, has been the manufacture of reliable springs, and when we consider the conditions under which such springs usually work, it will be seen that with any of the afore-mentioned indicators perfectly accurate results cannot well be obtained. Whenever an indicator is put in use, the piston quickly becomes

hot, and the heat is conducted to the spring. The spring is also heated by the steam which leaks past the piston; consequently, the spring rises to a much higher temperature than the surrounding atmosphere. If the pressure in the cylinder remained constant, the spring would ultimately assume the same temperature as the steam in the cylinder. If, however, the pressure varied rapidly, as in an ordinary steam engine, the transmission of heat would not be

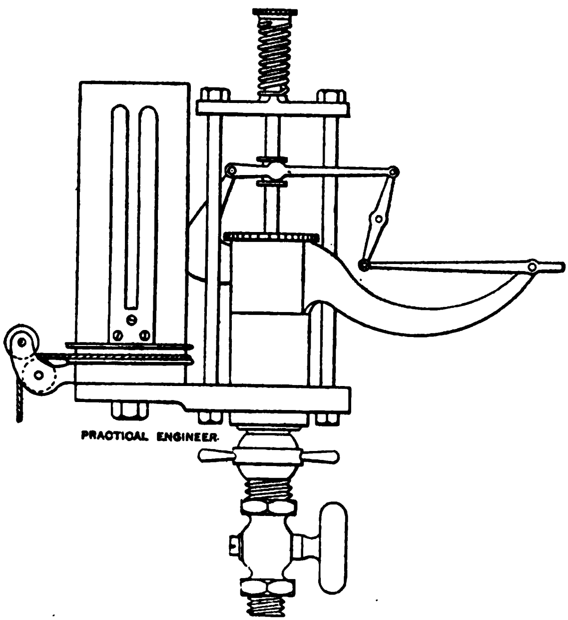


FIG. 13.—McKennell and Buchanan Indicator.

sufficiently rapid to permit of the spring temperature following closely the rapid changes of temperature of the steam.

Again, it is generally known that the strength of a spring varies with the temperature; hence, to ensure an accurate spring, the final tests should be made with the spring at the temperature which it will be at in actual service. This,

however, is impracticable, as, in the first place, its temperature is not known, and, secondly, it is liable to considerable variation. The best course would appear to be to test the spring throughout its scale, at the temperature corresponding to each pressure—that is to say, test it on an actual steam boiler against standard gauges or mercury columns, as it hardly seems likely that the error of a spring tested under such circumstances would then be great.

With a view of getting over this difficulty respecting springs, Messrs. McKinnell and Buchanan have patented a form of indicator in which the spring is quite away from the indicator piston and cylinder, and is well exposed to the atmosphere; consequently, the spring will generally be at about the same temperature as the atmosphere, and at the same time be readily accessible for changing. With such an arrangement the springs can be tested under exactly their working conditions, which, as will be seen from previous remarks, is desirable. This arrrangement will also facilitate the changing of the spring, as to do this it will not be necessary to touch the hot parts of the indicator. For external elevation of this indicator see fig. 13.

CHAPTER II.

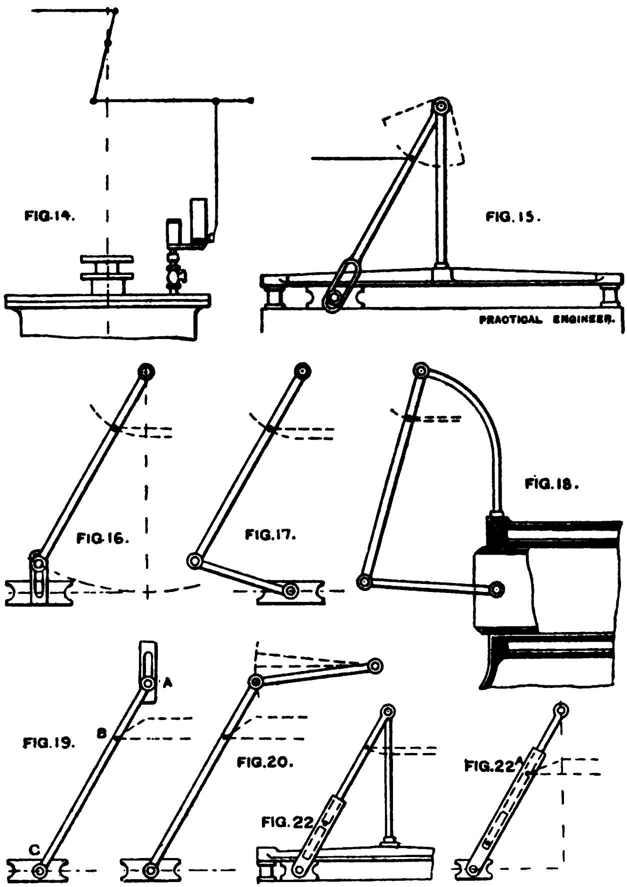
Reducing Gears for Indicating.

Turning from indicators to the means adopted for driving the indicator barrels, we find that these are even more numerous than the indicators themselves.

The first condition of a satisfactory indicator barrel driving arrangement is that the point to which the indicator cord is attached must move exactly in accordance with the engine piston, but to a reduced scale—that is to say, the point in question must be at each end of the stroke, at $\frac{1}{3}$, $\frac{1}{4}$, $\frac{1}{5}$, $\frac{1}{6}$, &c., of the stroke, exactly at the same time that the piston reaches these points, and similarly for every other position of the stroke.

An indicating gear should also be simple in construction, and not liable to get out of order or deranged in any way, and should be so arranged that the string may be led as directly as possible to the indicator, so as to have the string no longer than is absolutely necessary, thus lessening the risk of error through stretching of the cord.

For beam engines it is not often necessary to erect any special arrangement, as the string may be attached to the



parallel motion at such position that the desired travel will be given. This is shown at fig. 14. The indicator is usually attached almost directly to the top cover, whilst the connec-

tion to the bottom end of the cylinder is usually made by pipes to the sides of cylinder, or to the bottom, the pipes being arranged so that the indicator will stand vertical.

For horizontal engines an indicating gear consisting of a swinging or pendulum lever suspended at the top, while the bottom end is connected in some manner with the engine crosshead, is frequently adopted. Several forms of this arrangement are shown by the following illustrations.

Fig. 15 shows an arrangement which is of a very inexpensive character, and frequently is very easy of erection, especially when the engine-house is low, or when there is a beam fairly low down, suitable for carrying the pivot for the top end of lever. If neither of these conditions exist, a support might be carried from some convenient portion of one of the slide bars, or by arranging a pair of boards in the manner shown at fig. 21.

This arrangement does not fulfil all the conditions previously explained, as the motion of pin to which the cord is attached is not strictly coincident with that of the crosshead pin. The gear is, however, improved in this respect if the cord is attached to a disc fixed on the lever, as shown in dotted lines on fig. 15, the circumference of the disc being circular. Even with this alteration the gear would still not be quite accurate, and the amount of error would depend on the length of the lever, being greater in the case of a short lever than with a longer one.

The application of a disc with circular rim would, in addition to improving the gear as regards accuracy, render it suitable for leading the string in any desired direction. Where such a disc, or its equivalent, is not attached, the cord should always be led away from the driving point in a direction parallel to the centre line of the engine cylinder. The disc or pulley just described is usually termed a Brumbo pulley.

Fig. 16 shows a device which gives more accurate results. In this case a pin fixed in the lower end of the swinging lever slides in a vertical slide or slot connected to the engine crosshead. Of course, in this arrangement the slot must be exactly perpendicular to the line of motion of the crosshead. With such a gear the horizontal movement of the cord will be strictly proportional to the movement of the crosshead, but a source of error exists owing to the rising and falling motion given to the cord. If the distance between the swinging lever and the indicator is short, such error might be appreciable; but if fairly long it would be slight and

practically nil. A Brumbo pulley would render this gear less accurate, and should therefore not be attached.

The reducing gear shown at fig. 17 is a modification of that just described, the only change being the substitution of the radius link for the vertical slide. It has been very extensively adopted, and gives fair results, providing the radius link connecting the pendulum lever to the crosshead is fairly long; if, however, the link is short, material inaccuracy may arise. It is also desirable that the indicator cord be not very short, owing to the vertical movement of

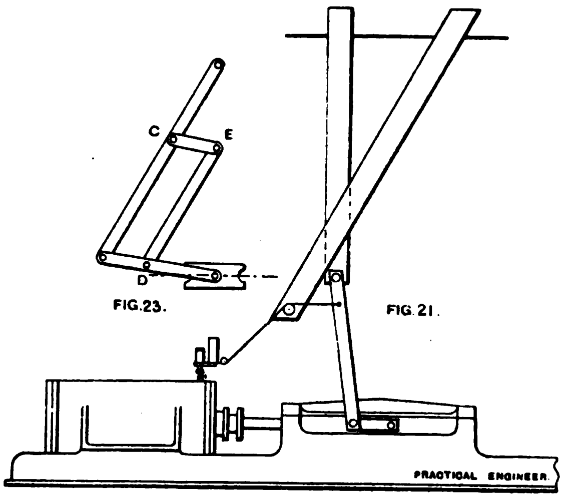


FIG. 23. FIG. 21.

the driving pin. A Brumbo pulley should not be connected to the swinging lever of this rig.

For gas engines the last-mentioned arrangement has been very generally adopted, the connections being frequently made in the manner shown by fig. 18, the support for the pendulum lever being a bent tube with its lower end screwed into or otherwise attached to the front end of the cylinder, while the radius link is attached direct to a small pin screwed into the side of the trunk piston.

A more uncommon device is that at fig. 19. In this case the upper end of the swinging lever moves in a slide, whilst the lower end is pivoted to the crosshead. The horizontal

movement of the cord pin will at all times be strictly proportionate to the movement of the engine piston, and will be in the proportion of A B : A C. There will, however, be considerable vertical motion, which will introduce a source of error, the amount of which will vary according to the length of the cord; that is to say, the error will be less if the cord is long than if it is short.

Sometimes the upper slide is substituted by a radius link, which is equivalent to substituting a rod of finite length for one of infinite length; hence the link should always be as long as possible.

A Brumbo pulley would lessen the accuracy of this gear.

Instead of making the pendulum with slides or radius links, it is often made in two parts, one sliding within the other, so as to allow for the varying distance between the crosshead pin and the upper pin, on which the swinging lever is pivoted. Usually this is done by making one portion of round iron rod, and the other of iron tubing.

With this form of indicator gear the cord should always be attached to the upper half of the arrangement, as per fig. 22, and not to the lower half (fig. 22A), as in the latter case the error is greater.

The two last-mentioned devices are practically identical in principle to fig. 15.

Turning from simple lever arrangements to those intended to give absolutely accurate results, it is probable that one or other of the various forms of pantograph will be the most frequently met with.

The form shown at fig. 23 is especially suitable for engines having a long stroke, and might easily be constructed from the pendulum gear at fig. 17, by adding the two links C E and E D. The motion of the indicator cord thus obtained is accurate in every respect, there being no vertical movement of the cord, and the horizontal movements being strictly proportional to the movements of the piston. This may be proved as follows :—

Let C and C_1 (fig. 24) be any two positions of crosshead pin, and let B D E F and B_1 D_1 E_1 F_1 be the positions of the various points corresponding to C and C_1 respectively. In the construction of the gear, D F is made parallel to A B, and E F to B C, and the lengths of these links are so proportioned that F is in the straight line drawn between A and C. Then

$$\frac{AF}{AC} = \frac{EF}{BC}$$

because E F is parallel to B C, and A E B and A F C are straight lines; now

$$E_1 F_1 = E F \text{ and } B_1 C_1 = B C$$

$$\therefore \frac{E_1 F_1}{B_1 C_1} = \frac{E F}{B C} = \frac{A E_1}{A B_1}$$

$$\therefore A F_1 C_1 \text{ is a straight line, and } \frac{A E_1}{A B_1} = \frac{A F_1}{A C_1}$$

We therefore now have

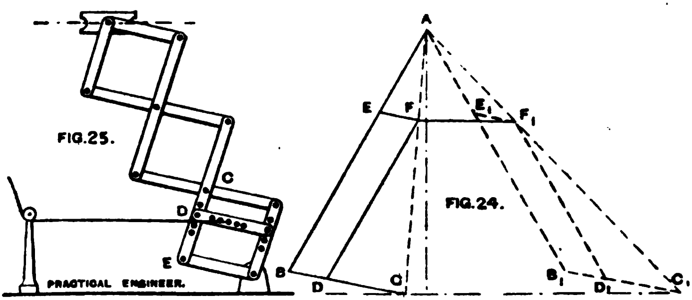
$$\frac{A F}{A C} = \frac{E F}{B C}$$

also

$$\frac{A F_1}{A C_1} = \frac{E F}{B C}$$

$$\therefore \frac{A F}{A C} = \frac{A F_1}{A C_1}$$

$$\therefore F F_1 \text{ is parallel to } C C_1.$$



But $F F_1$ represents the line of movement of the point to which the cord is attached; hence we have proved that this point moves parallel with the line of movement of the crosshead.

Again, because A F C and $A F_1 C_1$ are straight lines, and $F F_1$ is parallel to $C C_1$, it follows that

$$\frac{F F_1}{C C_1} = \frac{A F}{A C} = \frac{A E}{A B};$$

that is to say, the ratio of the movement of the cord pin to that of the crosshead is in the proportion of A E to A B throughout the stroke, as the above reasoning would apply to all other positions of the stroke.

This gear is not infrequently made of light baywood rods, with brass ferrules on the ends ; but light iron rods are rather preferable, as there is then no risk of the accuracy being affected by shrinkage of any of the rods. Of course it is essential that a guide pulley be fixed in such a position that the portion of cord leading away from the pantograph will be quite parallel with the line of motion of the crosshead—*i.e.*, with the centre line of the engine.

Another pantograph motion, similar to fig. 25, is sometimes used, and is an adaptation of the combination of levers known as the "Lazy tongs." This also gives quite accurate results, as may be proved in a similar manner to that just dealt with. A disadvantage of this rig is that there is a considerable number of joints, a little play at each of which might materially affect the results. It, however, has an advantage in that, by having a few spare holes in the lower section, as shown, the travel of the cord can be adjusted. Care, however, must be taken to see that the point to which the cord is attached is in a straight line, drawn through the centres of the pins connecting the various sections. The ratio of the travel of the cord to that of the piston will equal

$$\frac{\text{C D}}{\text{C E}} \times n$$

where n = the number of sections.

Two simple pulley-reducing arrangements are shown at figs. 26 and 27. In the former a cord passes round two pulleys, and has each end connected to the crosshead. The indicator cord is connected with a much smaller pulley fixed on the axle of one of the two larger ones. Not infrequently the axle itself is made the necessary diameter. In the latter, fig. 27, the cord connected with the crosshead only passes round one pulley, the return being effected by a coiled spring contained in a suitable case. Each of these arrangements gives accurate results at moderate speeds, but at high speeds the inertia of the pulleys becomes important, and may cause incorrect results.

It not infrequently happens that none of the forementioned gears can be applied to an engine—as, for instance, most of the high-speed enclosed engines.

In such cases quite special designs have to be occasionally adopted, but generally the indicator cord is actuated by means of a small crank, or eccentric fixed on the crank shaft. When either of these is used the length of the eccentric rod should be such that its ratio to the throw of

the eccentric or crank is the same as in the case of the engine connecting rod and crank, otherwise the motion of the indicator barrel will not coincide with that of the engine piston. It is also very necessary to see that the eccentric or crank is set in the proper position, otherwise the diagrams may be extremely inaccurate.

As regards the cord for actuating the indicator barrel, this should always be such that it will not be materially stretched by the pull which comes on it during work, and it is generally best to use cord specially woven for the purpose, and in the case of long lengths to use flexible wire. Ordinary cotton cord should never be used.

When connecting the cord up to an indicator it is almost always necessary to adjust its length several times, and in

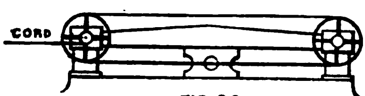


FIG. 26.

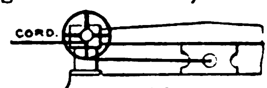


FIG 27.

FIG. 28.

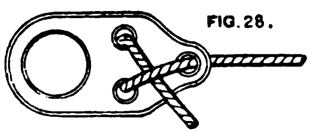


FIG. 29.

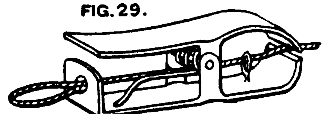


FIG. 30.

Practical Engineer.

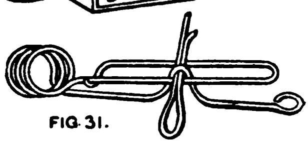


FIG. 31.

view of this most engineers prefer to use some device to facilitate such adjustment, instead of tying and untying knots.

A very large number of clips, &c., have been designed for this purpose, but those shown by figs. 28, 29, 30, and 31 give a general idea of the lines on which such devices are arranged. The form shown at fig. 28 is that supplied by the Globe Engineering Company Limited with the Tabor indicator, whilst that at fig. 29 is supplied by McInnes and Co. with the McInnes indicator. Each of these, but especially the last mentioned and that shown by fig. 31, is certainly very convenient of adjustment. For slow speeds an ordinary tie clip, as sold by hosiers, &c., at one penny, may be used in the same manner as fig. 29.

The most common way of arranging the indicator tap and connecting it with the interior of the cylinder is shown at fig. 32, A being a small boss cast on the cylinder, and for convenience of moulding usually carried parallel to the flange, as shown in plan.

The standard size for tapping the socket or upper end of the tap is $\frac{3}{4}$ Whitworth bolt thread. Wherever the taps are connected with the sides or ends of a cylinder, suitable bends should be fixed for the taps to screw into, as the indicator should stand vertical in all cases, so that the friction of the working parts may be reduced as far as possible.

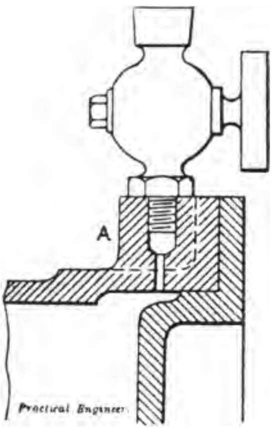


FIG. 32.

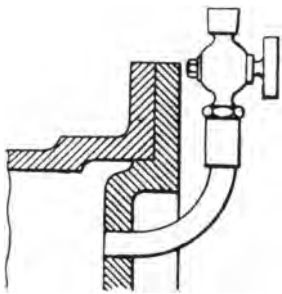


FIG. 33.

Frequently the opposite ends of a cylinder are connected by means of a copper loop pipe with a tap in the middle. Such an arrangement certainly lessens the trouble of indicating, but experience has shown that unless regularly used and cleaned out the pipe is very liable to get more or less choked with greasy deposit, and this frequently renders the indication worthless.

Diagrams illustrating this point will be given further on when dealing with defective indicator diagrams. For the reason just stated, indicator taps should be connected as directly as possibly to the openings leading into the cylinder, and these openings should be as straight and uniform as possible, and so near the end of the cylinder

that they will not be either wholly or partially covered by the piston at any point of its stroke, whilst at the same time there is a free passage to each opening for the steam.

As a rule, when an engine has not been arranged for the attachment of indicator taps, it is most convenient to connect these to the cylinder ends. In the case of horizontal engines, the method shown at fig. 33 is satisfactory.

The indicator tap opening should never lead into a steam port, as the rush of steam along such ports is liable to register wrong pressures on the diagram paper.

CHAPTER III.

We have seen in the previous matter how it is that the diagram produced on the indicator paper, if taken properly and with suitable appliances, represents accurately the pressure of steam at each different point of the stroke of the engine piston, and we now come to consider the various forms of the diagrams themselves.

In doing this we will, for the present, confine ourselves to diagrams relating to steam engines, and as it will be necessary to use the terms "stroke," "length of stroke," and "fraction of stroke" frequently, it would be well to define these at once.

The "stroke" of a piston is the course over which it travels in moving from one end of its cylinder to the other.

The "length of stroke" is the distance from end to end of the stroke, and is usually expressed in feet.

The "fraction of stroke" is the fraction which represents the position of the piston at any given instant: thus, if a piston having a stroke of 5 ft. stood 1 ft. away from one end of its stroke, the fraction of stroke would be $\frac{1}{5}$ or ·2, and so on for each other position.

Having now decided as to the meaning of these simple terms, we will proceed to consider the form of diagram which would be obtained under theoretical conditions from a non-condensing engine in which the steam is admitted throughout the whole length of each stroke, whilst the exhaust flows freely into the atmosphere during the full length of each return stroke.

To represent this by a diagram, we will begin by drawing the base or atmosphere line, and then marking A B, fig. 34, of such length that A B represents the length of stroke.

If the piston were in the position A, and steam at, say, 30 lb. pressure were suddenly admitted to one end of the cylinder without throttling, it would cause the pressure at that end to suddenly rise to 30 lb. and this would be represented on the diagram by a vertical line A C, $1\frac{1}{2}$ in. high, taking a vertical scale of 1 in. = 20 lb., or $\frac{1}{20}$ in. = 1 lb.

If now the steam continued to press on the piston with a force of 30 lb. per square inch throughout the stroke, this would be shown on the diagram by C D parallel to A B. At D the pressure suddenly falls to atmospheric, owing to the exhaust port being opened. This is shown by the vertical line D B; then during the return stroke the line on diagram will correspond with A B, the atmospheric line, as the exhaust is supposed to pass away freely.

From the foregoing we see that the diagram representing the action of steam in the cylinder of an engine working

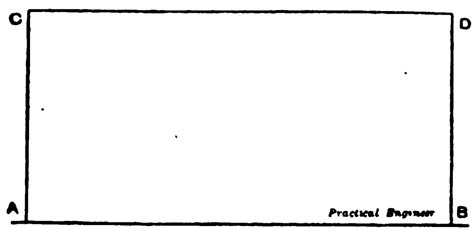


FIG. 34.—Theoretical diagram of non-condensing engine. No back pressure, and steam carried full length of stroke.

under the described conditions would be a simple rectangle in form. The conditions as regards the periods of steam admission and exhaust under which the engine in the example just taken was supposed to work are not infrequently met with in practice, but those relating to the admission of steam without drop of pressure and exhaust at atmospheric pressure are not, consequently, actual indicator diagrams, generally differ from fig. 34 at the steam and exhaust lines C D and A B. Fig. 35 shows such differences as might be expected. The sloping upper line in this latter is due mainly to the steam on its way to the cylinder being throttled, or reduced in pressure, in passing through the throttle valve, ports, &c., whilst the space between the lower line of the diagram proper and the atmospheric line is caused by throttling in the exhaust ports and pipes.

The next type of diagram we will consider is that for an engine in which the pressure on the piston is not intended

to be uniform throughout the stroke, as would be the case

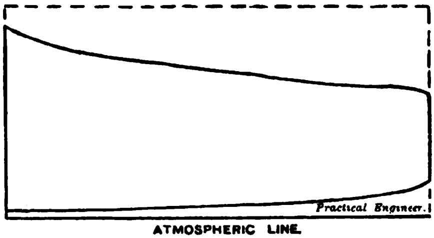


FIG. 35.—Probable modification of fig. 34, caused by the throttling of the steam and exhaust ports.

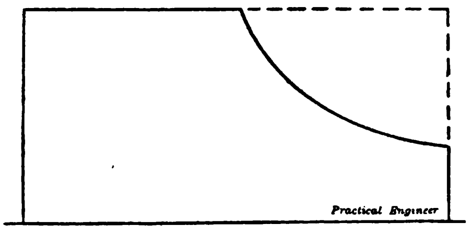


FIG. 36.—Theoretical diagram from non-condensing engine; steam cut off at half-stroke.

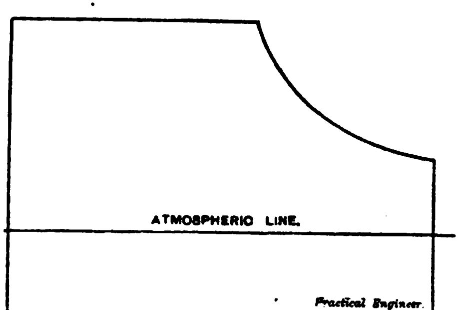


FIG. 37.—Theoretical diagram, same as fig. 36, with addition of condenser.

if the steam supply to the cylinder were cut off while the

piston had still some portion of its stroke to travel. After the steam was cut off, the pressure on the piston would gradually decrease as the piston advanced. The rate at which this reduction would occur will be considered later on.

The diagram shown at fig. 36 represents the theoretical diagram for these conditions, supposing that the cut-off occurs at half stroke. The difference between this and fig. 34 is that the latter half of the upper line—that is, the portion representing the pressures after the steam supply to the cylinder is cut off—is curved downwards, so as to represent the gradual drop of pressure, instead of being horizontal. The remaining parts are identical. The differences between this and diagrams taken from actual engines of this class will receive attention further on.

If in either of the before-mentioned engines the exhaust were passed into a suitable condenser instead of direct to the atmosphere, the lines representing the pressures acting against the piston during its return strokes would be below the atmospheric line, the distance below varying according to the vacuum obtained.

Fig. 37 shows fig. 36 modified in this respect. In drawing the two last-mentioned theoretical diagrams, the only difficulty likely to be felt will be at the curved portions or expansion curves, as they are usually named, and with a view of drawing these intelligently, at the same time dealing with matter which will be valuable in the further consideration of indicated diagrams, it will probably not be out of place to consider very briefly a few of the principal properties of steam.

If heat were applied to water in an open vessel, the temperature of the water would at once commence to rise, and this would continue until the water began to boil, after which no increase of temperature would occur, the whole of the heat supplied being utilised in transforming the water into steam; and the temperature to which water under such conditions could be raised may be taken as 212 deg. Fah.

If, instead of being in an open vessel, the water were enclosed, the temperature at which it would boil would be dependent on the pressure in the vessel: if higher than the ordinary atmospheric pressure, the boiling point would be higher; but if lower, the boiling point would be lower.

In all cases, whatever the pressure on the liquid, the temperature will not rise above the boiling point until all the water has been changed to steam, providing the pressure

remains constant; the heat added to the water after ebullition has commenced being, as in the former instance, utilised in evaporating and not in raising the temperature. Owing to its apparent disappearance, such heat is usually termed latent heat, while that added during the period that rise of temperature is observed is called sensible heat.

A table giving boiling temperatures for various pressures, and several other properties of steam, is given at the end of the book.

Steam such as we have just considered—that is, steam at the pressure and temperature at which it was generated—is called "saturated steam." It is this class of steam which we have to deal with in almost all steam-engine diagrams.

The connection between the temperature and pressure in the saturated steam just described and referred to in the tables mentioned is very important, and must always be borne in mind when studying indicator diagrams; for instance, the tables show that at a temperature of 141·6 deg. Fah. water boils when the pressure on the surface falls to 3 lb. per square inch. This being the case, it is easy to see that so long as a vessel contained water at this temperature the pressure on the surface could not possibly be reduced to less than 3 lb. per square inch; consequently, if the water in a condenser were at 140 deg. Fah., the best vacuum possibly obtainable would be 3 lb. absolute pressure, or, if atmospheric pressure were 14·7 lb. per square inch, the best vacuum would be $14·7 - 3 = 11·7$ lb. per square inch.

With a temperature in the condenser of, say, 120 deg. Fah., the best possible vacuum would be about 12·8 lb.

Other instances showing the usefulness of saturated steam tables will be met with further on.

When steam is raised to a higher temperature than that corresponding to its pressure, it is called superheated steam. Steam in this condition is not so frequently met with in engine practice, but it is worth while to note that superheated steam must necessarily be quite dry—that is, must be quite free from water—as, even if water were by any means introduced into it, the water would at once boil away to steam, owing to the temperature being above the boiling point corresponding to the pressure.

The steam used in a steam engine does not during expansion follow any known law, but for our present purpose, in considering the changes of pressure which occur during expansion, we may suppose that it acts as a perfect gas, and

remains at the same temperature throughout the expansion, as Boyle's well-known and simple law can then be used.

Boyle's law states that "the pressure of a gas is inversely proportional to its volume during compression and expansion, providing the temperature remains constant."

NOTE.—The absolute pressure must always be taken—that is, the pressure above atmosphere plus 15, the approximate atmospheric pressure.

As an illustration of the application of this law, take the case of a steam engine in which the steam is cut off at, say, one-tenth of the stroke, the initial pressure being 85 lb. above the atmosphere. Let the sectional area of the cylinder be A and the length of stroke S.

Then at the point of cut-off the volume of steam in the cylinder will be A × ·1 S = ·1 A S. At $\frac{2}{10}$ths (·2) of the stroke the volume will be A × ·2 S = ·2 A S—that is, twice as great as at the point of cut-off. Then, as the pressure is inversely proportional to the volume, the pressure at ·2 of the stroke will be half as high as at ·1 = $\frac{100}{2}$ = 50 lb. per square inch, 100 lb. per square inch being the absolute pressure.

At ·3 of	stroke the	pressure	will be	$\frac{100}{3} = 33\frac{1}{3}$	lb. per sq. in.
·4	,,	,,	,,	$\frac{100}{4} = 25$	,,
·5	,,	,,	,,	$\frac{100}{5} = 20$	,,
·6	,,	,,	,,	$\frac{100}{6} = 16\frac{2}{3}$	,,
·7	,,	,,	,,	$\frac{100}{7} = 14\frac{2}{7}$	,,
·8	,,	,,	,,	$\frac{100}{8} = 12\frac{1}{2}$	,,
·9	,,	,,	,,	$\frac{100}{9} = 11\frac{1}{9}$	,,
·10	,,	,,	,,	$\frac{100}{10} = 10$	,,

The same result might be arrived at in a slightly different manner by stating Boyle's law, as follows:—

With a perfect gas the product obtained by multiplying the pressure during expansion or compression by the volume is a constant quantity, if the temperature does not vary. $\therefore$ $p\,v$ = constant, where p = the pressure at any volume v.

In the example just taken with the cut-off at one-tenth of the stroke, $p\,v = 100 \times \cdot 1 = 10$; $\therefore$ throughout this expansion $p \times v$ must = 10.

At ·2 .. .. $p \times \cdot 2 = 10$; $\therefore$ $p = \frac{10}{\cdot 2} = 50$;

At ·3 .... $p \times \cdot 3 = 10$; $\therefore$ $p = \frac{10}{\cdot 3} = 33\frac{1}{3}$,

and so on to the end of the stroke.

From the foregoing it will be seen the pressure at any point during the expansion of steam may be easily determined approximately.

The next process is to construct a theoretical indicator diagram, supposing the engine to be of single-cylinder condensing type. Let A B, fig. 38, represent the base or absolute zero line of the diagram. Divide this into ten parts, then at A mark off the initial pressure of 100 lb. per square inch to any desired scale. At ·1 mark the pressure of 100; at ·2, 50; at ·3, 33⅓; at ·4, 25; and so on till the last point is reached. Then draw the curve D E through the points thus obtained, afterwards completing the diagram in the manner previously described.

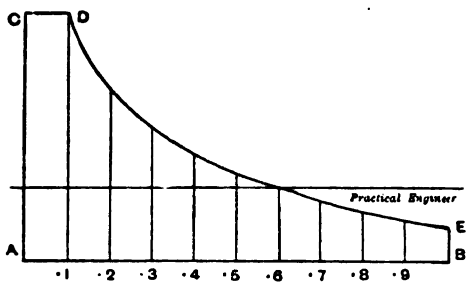


FIG. 38.—Construction of theoretical diagram.

The curve thus obtained will be a hyperbolic curve; hence, if desired, purely geometrical methods may be adopted in its construction. As a rule the geometrical methods are preferable as being simpler, especially when the point of cut-off is at some odd fraction of the stroke, or when clearance is taken into account, as should always be the case.

The clearance in a steam-engine cylinder is the cubic contents of the space between the piston and the valve or valves when the piston is at the end of the stroke. As this space is occupied by steam, it is evident that the volume of steam which should be reckoned on when drawing an expansion curve is that in the cylinder proper, plus that in the clearance space. To facilitate this the clearance volume should be expressed as a percentage of the volume swept through by the piston.

For ordinary cylinders the average clearance may be taken as from 4 to 6 per cent.

With a view of illustrating what is probably the simplest geometrical method of drawing hyperbolic curves, we will take the case of a steam engine in which the steam is cut off at, say, one-sixth of the stroke.

Draw A C, fig. 39, to represent the length of stroke, and mark point B so that ratio of A B to A C = $\frac{1}{6}$—that is, fraction of stroke at which steam is cut off. Then make A X so that

$$\frac{A\,X}{A\,C} = \frac{\text{clearance}}{\text{volume swept through by piston}}.$$

Then erect vertical lines X Y, A D, C F, and draw Y D F, the height of the vertical lines being such that they represent to some known scale the absolute pressure in the cylinder at the point of cut-off. Divide D F into any number of parts, not necessarily equal, as shown, and draw a line from each to the corner X, each of these to intersect B E. Where each diagonal intersects E B, draw horizontal to cut the vertical line, from top of which the diagonal line was drawn. The intersection gives one point in the hyperbolic curve; any number of other points may be similarly obtained.

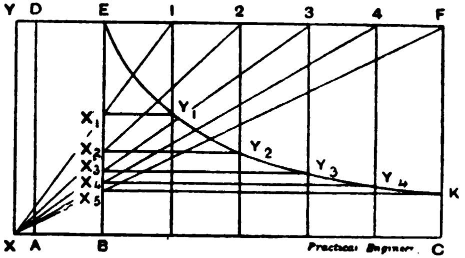


FIG. 39.

Where many curves have to be drawn, it is very convenient to have a number of hyperbolic curves drawn on tracing paper, and then simply place the tracing in suitable position, and prick a number of points through the curve desired; or, what is still handier, place the tracing on a well-lighted window, and then adjust the diagram over it to the proper position, afterwards tracing the required curve direct by pencil, as there will generally be sufficient transmitted light for this to be done without difficulty. This system is also very convenient for drawing hyperbolic curves on actual diagrams for the purpose of comparison. It

is also by far the most convenient when other curves, such as adiabatic or saturated, are desired, the only material trouble being the making of the original tracing. The last-mentioned trouble is not, however, necessary in the case of hyperbolic curves, as these can be obtained, drawn on stout tracing paper, in a very neat form, and at a low cost, from the makers of the Tabor indicator.

We have now seen how to draw simple theoretical diagrams from condensing and non-condensing single engines, both when the steam is used expansively and when it is not. The next step will be to ascertain in what manner and to what extent the diagrams taken from ordinary engines, with valves well set and working under favourable conditions, would differ from these.

Starting with the steam line (C D, fig. 38), we shall always find that this is a few pounds below the boiler pressure. A little consideration will show that this must necessarily be the case, for if the pressure in the boiler and cylinder were quite equal, there would be no force available for setting into motion the mass of steam in the pipes, &c.; hence, there must always be some difference, and the higher the speed at which it is necessary for the steam to move through the pipes, the greater will be the difference in the pressures; consequently, the pipes should be of such diameter that the rate of flow through them may not be excessive. In addition to causing greater drop of pressure in the manner just stated, a high velocity of steam would also cause increased friction in pipes, and it is probable that the losses through friction are the greatest of any. It must, however, be borne in mind that any increase in the diameter of pipes exposes a greater surface to the cooling action of the surrounding atmosphere; hence, in trying to reduce the frictional losses, care must be taken, or the condensation resulting from the increased surface may more than counterbalance the gain derived. When the piston of an engine is approaching either end of its stroke, and just before the valve opens to admit steam into the cylinder the steam in the pipes will be practically stationary, but immediately the valve opens and the piston commences to move forward the mass of steam in the pipes will require setting in motion, and its velocity will be gradually increased until the point of cut-off is reached, or, if this is later than half stroke, until the piston about reaches its mid position. In consequence of this, the pressure at the piston end will fall until the pressure at the opposite end is sufficiently in excess to

produce in the mass of steam the required velocity; hence the difference of pressure due to this cause will be proportional to the weight of steam in the pipes, which will in turn be almost in direct proportion to the pressure of the steam and the length of the pipes.

Again, the speed at which the engine runs will affect the drop of pressure. To illustrate this. we will compare two engines, one running at, say, 30 revolutions per minute, the other at 120, and to simplify matters we will suppose that in each case the pipes are of such diameter that the average speed of flow through them is 100 ft. per second, and that the cut-off in each engine occurs at half stroke.

Under these circumstances the maximum velocity will be nearly 1·57 times the mean velocity.

The engine running at 30 revolutions per minute makes one revolution—*i.e.*, two strokes—in two seconds, or half a stroke in half a second; consequently the velocity of the steam in the pipes is raised from nothing to 157 ft. per second in half a second. Therefore the acceleration per second is $157 \times 2 = 314$ ft.

In the case of the engine running at 120 revolutions per minute, it will readily be seen that the steam velocity of 157ft. per second has to be generated in one-eighth of a second; consequently the acceleration is now at the rate of 1,256 ft. per second, or four times as great as with the slower engine. From the laws relating to moving bodies we know that the rate of acceleration is proportional to the forces applied; hence the difference of pressure required to give the steam the necessary motion will be four times as great in the quicker-running engine than in the slower one, and, generally speaking, will be proportional to the number of revolutions per minute.

From the foregoing remarks we see that there are several things tending to cause loss of pressure between the boiler and the engine, so that when arranging steam piping it becomes necessary to treat each case on its merits. A general rule is to proportion the pipe so that the average rate of flow of the steam may be about 80 ft. per second; it would, however, appear better to alter this number slightly to suit the circumstances of the case. Possibly velocities such as the following would give good all-round results:

						Feet per sec.
Low pressure	and	low number	of revolutions		......	100
Moderate	„	moderate	„	„	......	90
High	„	„	„	„	......	80
High	„	high	„	„	......	70

Of course in all cases steam pipes should be covered by some non-conducting material, and should be as little exposed to the outside atmosphere as possible; also they should be quite free from sharp corners. Wherever bends are necessary, these should be of large radius.

The next point to note is, that the change from the steam line to the expansion curve is usually not abrupt, as is shown on the theoretical diagrams, but is somewhat rounded, as shown by fig. 39. This is owing to the gradual closing of the steam port, as this causes the area of the opening for steam to be extremely small just prior to the steam being cut off; consequently considerable throttling occurs. Such throttling is frequently termed "wire-drawing." To reduce the loss from this source, and at the same time give other advantages, many valve gears have been introduced in which the steam valves are so arranged that they are closed very rapidly by means of springs or weights, instead of being closed slowly by a positive motion derived from the eccentrics; consequently "wire-drawing," such as has just been referred to, is reduced to a minimum—in fact, with such gears, it is frequently barely perceptible on the diagrams.

The wire-drawing before referred to frequently makes it difficult to judge from a diagram at what point the cut-off occurs, but as a general thing it may be said to be at that point where the curve changes from convex to concave.

Passing on to the expansion curves, it will generally be found that these do not correspond at all closely to hyperbolic curves, even when the valves and piston are in excellent order, and especially is this the case when the cut-off occurs at an early period of the stroke; in fact, it will not infrequently happen that a close approximation of the expansion curve to a hyperbolic curve indicates that considerable leakage is going on past the piston.

This result is brought about by several causes, the principal of which are liquefaction of steam during its admission to the cylinder and re-evaporation during expansion of the whole, or at least part, of the water of condensation thus formed. It therefore now becomes necessary to look briefly into these causes.

Dr. Joule showed, as the result of his well-known investigations between 1840 and 1849, that heat and work are mutually convertible, and that 1 unit of heat is equal to 772 foot-pounds of work; hence, in obtaining work from the expansion of steam or any other gas, the working fluid

must necessarily lose heat proportionate to the work done, and for every 772 foot-pounds of work performed 1 unit of heat will disappear, causing a lowering of the temperature of the steam or gas.

If, therefore, we were to admit steam to the cylinder of a steam engine for a portion of the stroke, and then allow it by its expansion to push the piston forward during the remainder of the stroke, performing during expansion an amount of work W, then $\frac{W}{772}$ units of heat would disappear during expansion, and consequently, if the expansion were

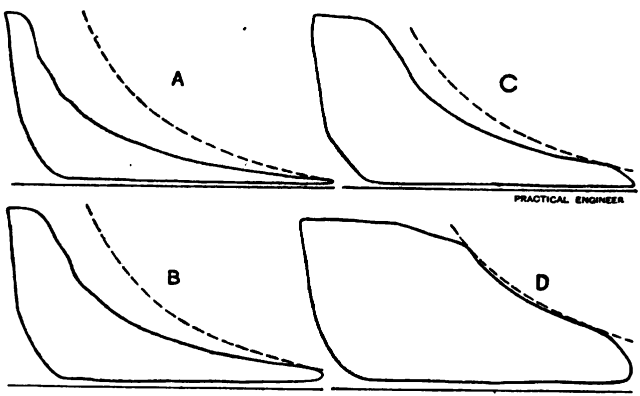


FIG. 40.—Diagrams showing how expansion curves vary when compared with hyperbolic curves.

considerable, the temperature of the steam would be materially lowered, and there would be a corresponding lowering in the temperature of the surfaces of the cylinder walls.

From this we see that the next steam which enters the cylinder will come into contact with surfaces cooler than itself, with the result that a portion of the incoming steam will be condensed to water, and this condensation will go on until the walls are of the same temperature as the steam.

It is not necessary for us to examine this question so closely as to ascertain the exact point of the stroke at which such condensation would cease to take place; as for

our present purpose it is sufficient to know that it would at least go on until the point of cut-off were reached. The weight of steam condensed after this point we will neglect, as its quantity, as a rule, will be very small compared with the weight previously condensed.

In consequence of the condensation just described, there will always be at the point of cut-off a greater or less quantity of water, as well as steam, and it is fair to assume that this water is at the same temperature as the steam it is in contact with. When the expansion commences, the pressure, of course, at once begins to fall, and, owing to the work performed, the temperature also falls; and this being so, the steam will, before long, fall to a lower temperature than the cylinder walls, which will, therefore, then commence to give out the heat they previously absorbed. Hence, the water of condensation previously formed will receive some of this heat and be re-evaporated into steam, thus causing rise of pressure, and this would go on until the end of the stroke is reached or the whole of the water changed into steam; the latter, however, will rarely occur before the completion of the stroke.

The amount of steam condensed during the admission of steam and the amount of water re-evaporated during expansion will vary according to the point of cut-off. The reason for this is obvious, when we remember that the cooling of the cylinder which caused the condensation was brought about by the heat transformed into work during expansion; consequently, the more prolonged the expansion, the greater will be the transfer of heat from the cylinder walls, and the greater the condensation and re-evaporation.

From the foregoing it will be seen that the expansion curve on an actual indicator diagram cannot, as a rule, be expected to agree closely with a hyperbolic curve. It will, however, be found that there will frequently be a fairly close approximation when the cut-off takes place late in the stroke, or when the cylinder is steam-jacketed.

Owing to the variations in actual steam expansion curves, some experience is required in forming reliable conclusions from the comparisons of such curves with hyperbolic, saturated, or other curves. As an instance of this the diagrams at fig. 40 are introduced. These are taken from a horizontal non-condensing engine, fitted with an automatic cut-off valve gear.

From these it will be seen that with the late cut-off (diagram D) the hyperbolic and the expansion curves agree

fairly closely, but that the difference between them gradually increases as the cut-off gets earlier.

The losses which this difference shows will be dealt with further on.

Passing on to the exhaust opening, it will be remembered that in the theoretical diagrams this was taken as occurring just when the piston had completed its outstroke. In practice, however, it is found desirable to open the exhaust port rather earlier, as will be seen from the following :—

By opening the exhaust port before the end of the stroke the steam is allowed to escape more freely from the cylinder, as the port is then more fully opened during the period at which the motion of the piston is comparatively slow, viz., near the ends of its stroke. The consequence is that the back pressure falls more rapidly than would otherwise be the case; hence the economy of the engine is improved. Instances of this are shown at fig. 41, in which *a* shows the usual form of the toe of the diagram when the

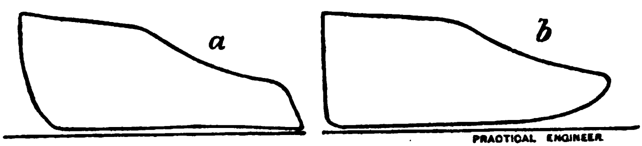


FIG. 41.—Diagrams showing effect of different points of exhaust opening.

exhaust open is fairly early; *b* shows the effect of late exhaust opening in retarding the fall in the back pressure.

A further and perhaps more important reason for early exhaust opening is that the pressure tending to press the piston forward is by this means reduced just at the end of the stroke, where it is desirable it be reduced, for the reason stated a little further on.

It is also usual to arrange the valves to close the exhaust port before the end of the stroke, with the result that the whole of the steam is not discharged from the cylind r at each stroke, and the portion which is in the cylinder at the time the exhaust port closes is compressed into a gradually decreasing space, until at the end of the stroke its volume is equal to that of the clearance volume of the cylinder.

This period of the piston's stroke is termed the "compression" or "cushioning" period, and the line on the diagram which shows the rise of pressure during such period is called the "compression curve."

To understand the benefits of early exhaust opening and closing, it is necessary to know a little about the forces brought into play by the reciprocating motion of the piston and its connections, such as the piston rod, crosshead, and connecting rod, and in considering these we will take for example an engine of the horizontal type.

At the instant the piston is at either end of its stroke the reciprocating parts are momentarily at rest. They, however, are at once set in motion, either by the pull of the crank pin or the pressure of steam in the cylinder, and their motion is gradually increased until the maximum velocity is reached, which would occur at half-stroke if the slight difference due to the angularity of the connecting rod be neglected. After the piston has reached mid-stroke, it commences to fall in speed until it again comes to rest at the opposite end of the stroko

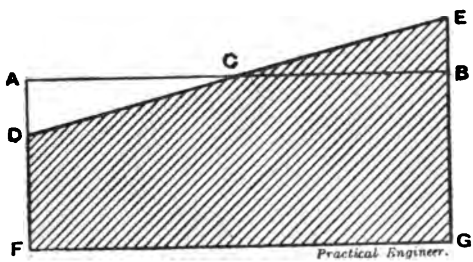


FIG. 42.—Diagram showing effect of reciprocating parts when speed is low and engine working without expansion. The shaded portion represents the actual pressures on the crank pin.

In order to produce the acceleration during the first half of the stroke it is necessary that a considerable force be applied, and it can be proved quite easily that this force is greatest when the piston is at the end of the stroke, and gradually falls until at mid-stroke—that is, the point where the maximum velocity is reached—the force necessary for acceleration disappears altogether, the magnitude of the force being proportional to the distance of the piston from the centre.

After mid-stroke the piston, piston rod, &c., would tend to move onwards at the maximum velocity, but cannot, of course, do so, as they are connected to the crank pin, and as these parts are brought to rest during the latter half of the stroke it follows that considerable pressure must be

brought to bear on the crank pin, in addition to that due to the steam pressure; and this additional force will be a gradually increasing one as the piston approaches the end of its travel, and, further, it will equal the force which was required to be put on the piston at each corresponding position during the accelerating period.

To determine these forces, the following formula, taken from Mr. A. Rigg's book on the steam engine, may be used:

$$\left.\begin{array}{l}\text{The pressure per square inch}\\ \text{at commencement of stroke}\\ \text{required to give the neces-}\\ \text{sary acceleration to the}\\ \text{reciprocating parts}\end{array}\right\} = p = \frac{\cdot 00034 \times R \times W \times r}{A}$$

R = revolutions per minute;
W = weight in lbs. of reciprocating parts;
r = radius of crank in feet;
A = area of cylinder in square inches.

Suppose that in a given case the pressure p, calculated from the above formula, were 10 lb., and that the line A B, fig. 42, represents the length of stroke, with C for the central position, draw A D to represent to any given scale 10 lb. per square inch, and afterwards draw straight line D C E through C; then B E will equal A D, and the vertical distance from the line A C at any point D C will represent the required accelerating pressure at that point, while the distances from C B to C E will represent the pressures per square inch brought to bear owing to the momentum of the moving parts.

From the above it will be seen that the actual effective pressure on the crank pin during the first half of the stroke is that shown by the indicator diagram, minus that required for acceleration of the moving parts, and as shown by the line D C. During the second half of the stroke the pressure on the crank pin will be that shown by the diagram, plus that shown by the line C E. In this example the rectangle A F G B represents the indicator diagram.

Fig. 42 and fig. 43 illustrate very clearly the foregoing remarks, and at the same time show the effect of increased speed.

In fig. 41 the pressure p calculated from the formula just given is taken as 10 lb., while in fig. 42 it is taken as 30 lb., the speed having been increased 1·7 times, say from 50 to 85. Owing to this increase of speed the pressure required to

overcome the inertia of the reciprocating parts at the commencement of the stroke practically equals the steam pressure, with the result that only a small portion of the work done by the engine is done during the first half of the stroke, and owing to the excessive pressure at the end of the stroke, the engine could not be expected to work at all smoothly.

If now the engine were supplied with steam at a higher pressure, and the cut-off were made to take place at say one-fifth of the stroke, the speed being the same as in the case of fig. 43, the actual pressures on the crank pin would be much more equal throughout the stroke than is usually considered. These pressures are shown by the

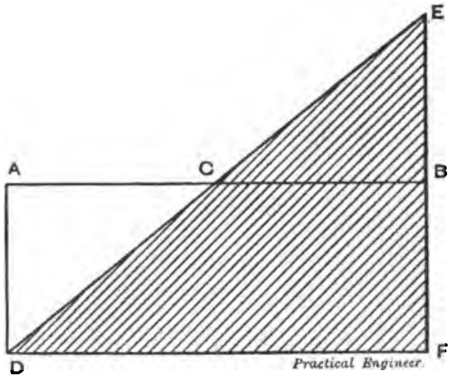


FIG. 43.—Diagram illustrating same as fig. 42, but with speed increased.

shaded area of fig. 44; the upper line of this area having been obtained by deducting the pressures required for acceleration during the first half, and adding the pressures due to momentum during the second half to the steam pressures shown by the diagram.

By applying the method just described, it will be found that by suitably proportioning the speed of the engine and the ratio of expansion, it is possible under all ordinary conditions to get the pressure on the crank pin fairly uniform throughout the stroke. To get smooth running, it is, however, desirable that when the crank is passing its centres—which is the point at which the pressures at the connecting rod and other joints change direction—the

pressure on the joints be as low as possible, so as to lessen the tendency to knock; hence the pressures tending to carry the piston forward, and those tending to retard its action, should be about equal when the crank is just approaching each centre.

For the reasons which have just been given, the pressures tending to carry the piston forward when near the end of the stroke are, that due to its momentum, together with the steam pressure. This being the case, it is desirable that a

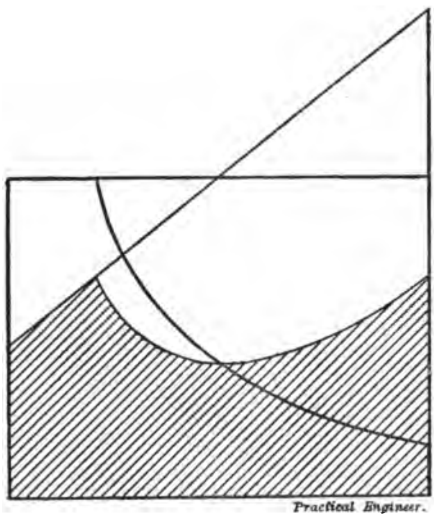


FIG. 44.—Diagram showing effect of reciprocating parts on engine working expansively.

force be applied on the opposite side of the piston equal to the two just mentioned, and this force is usually obtained by means of compression. Again, seeing that at the commencement of the stroke the actual pressure on the crank pin is much less than the total effective pressure on the piston, owing to the inertia of the moving parts, the pressure which would retard the motion of the piston should be as low as possible, and to effect this the exhaust port should be opened fairly early.

From what has been said, it will be seen that for smooth running the exhaust should open early, and be closed at such a point that the compression pressures at the end of the stroke may be about equal to the pressure on the opposite side, plus the pressure per square corresponding to the momentum of the piston.

Fig. 45 shows an indicator diagram which would be favourable to smooth running when the weight of the reciprocating parts and the speed of the engine were such

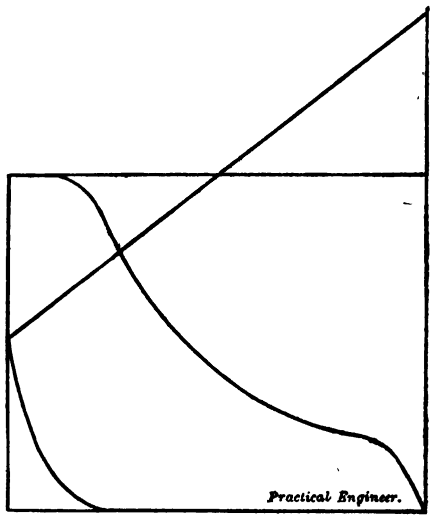


FIG. 45.—Diagram favourable to Smooth Running.

as to require an accelerating force of 30 lb. per square inch at the commencement of the stroke. The reason for this will be seen by referring to fig. 46, which shows the actual pressures on the crank pin.

As an instance of what class of diagram might be expected to give very bad running, fig. 47 is introduced. In this case the eccentric is set very late. At fig. 48 is the diagram of pressure on the crank pin, drawn on the assumption that the speed and the weight of the reciprocating parts remain

the same as in the previous example. For further information on this interesting portion of our subject, the reader is referred to Mr. A. Rigg's book on "The Steam Engine."

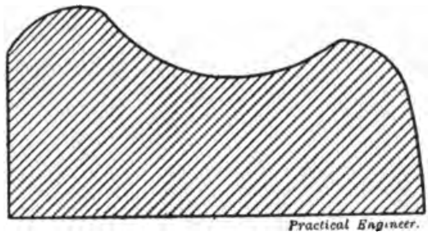


FIG. 46.—Diagram showing how Pressure on Crank Pin falls to nil just at end of stroke, with Indicator Diagram similar to Fig. 45.

Apart from its usefulness in tending to cause smooth running, compression is desirable, as within limits it is conducive towards economy. Possibly the principal reason for this will be found in the fact that the work done on the

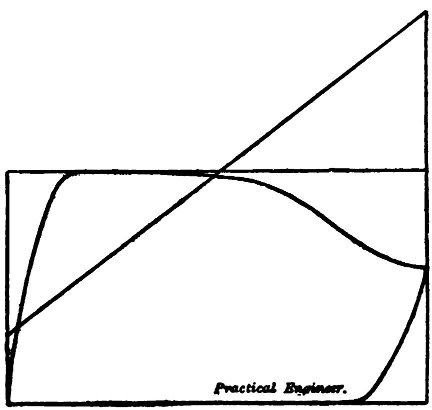


FIG. 47.—Bad Diagram for Smooth Working.

steam during compression causes the temperature of the steam to rise; hence on the opening of the steam ports there would in all probability be less condensation of the entering steam. This being the case, it would appear that the greater

the range of temperature in the cylinder, the greater should be the compression; hence rather more compression is desirable when the cut-off is early than when it is late.

Speaking generally, for ordinary engines the compression will, as a rule, not be far wrong if it is sufficient to meet the requirements already described for smooth running, and when we consider that in by far the majority of cases smoothness of running is of more importance than a slight gain economically, it will be seen that this amount of compression is probably the best to aim at.

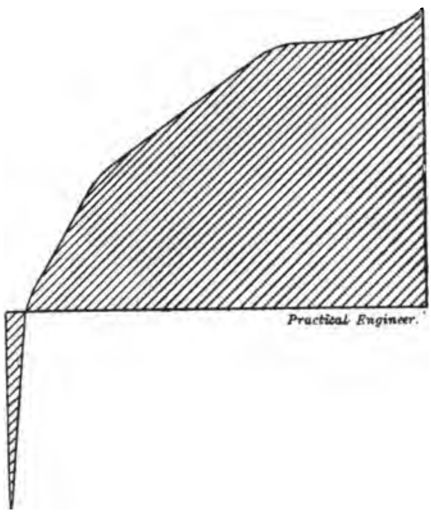


FIG. 48.—Diagram showing great Change of Pressure which occurs at end of stroke, when Indicator Diagram is similar to Fig. 47.

The rapidity at which the pressure rises during compression depends on the amount of clearance in the cylinder. If the clearance is little, the pressure will rise rapidly; but if considerable, then the pressure will rise comparatively slowly; hence, when determining the points at which the exhaust port should be closed by the valve, the volume of clearance should be borne in mind, so that the desired pressure may be reached at the end of the stroke.

Up to now we have only considered indicator diagrams as showing the distribution and action of the steam in the

cylinder. They, however, have another very important function, as by their means the actual amount of work which is being performed by the engine can be ascertained.

It is customary to express this work in "horse power." Watt is said to have originated this term, as he found it necessary when putting down or tendering for one of his engines to state how many horses' work it would be capable of performing, and in consequence of this he made experiments to ascertain the average amount of work per minute performed by a horse, with the result that he decided that the term horse power, usually written H.P. or HP., should represent 33,000 foot-pounds of work per minute, and this value, although known to be high, has since been retained.

In a steam engine the amount of work done in one stroke in foot-pounds is equal to the average effective pressure on the whole of the piston in pounds, multiplied by the length of stroke in feet; the pressure on the whole of the piston being equal to the average pressure per square inch, multiplied by the area of the piston in square inches.

Let P = pressure per square inch in pounds;
A = sectional area of cylinder in square inches;
L = length of stroke in feet;
N = number of strokes per minute;

then $P \times A \times L$ = work done in one stroke;

$\therefore P \times A \times L \times N$ = work done in N strokes = work done per minute.

But we know that 1 horse power is taken as 33,000 foot-pounds per minute; therefore the work done per minute, divided by 33,000, will give the horse power.

$$\therefore \text{H.P.} = \frac{P.A.L.N}{33000}$$

or, for the purpose of aiding the memory, the letters in the numerator may be re-arranged, so as to form the word PLAN. Thus—

$$\text{H.P.} = \frac{P.L.A.N}{33000}.$$

In those cases where it is necessary to frequently calculate the I.H.P. of any particular engine, the process may be simplified by booking a constant for each speed, which, when multiplied by the average pressure, gives the I.H.P.

The constant is obtained from the preceding formula by simply separating P from the remaining quantities, thus :

$$\text{I.H.P.} = \frac{P \cdot L \cdot A \cdot N}{33000}$$
$$= P \times \frac{L \cdot A \cdot N}{33000}$$
$$= P \times \text{constant},$$

the constant being the calculated value of

$$\frac{L \cdot A \cdot N}{33000}.$$

For any particular engine L and A remain unchanged, whilst N is not, as a rule, liable to much change ; hence a few constants, calculated as above, will generally suffice.

It may also be noted that $L \times N$ is equal to the piston speed ; therefore, if $S =$ the piston speed,

$$\text{I.H.P.} = \frac{P \cdot A \cdot S}{33000}.$$

If N is taken as the number of strokes per minute during which pressure is exerted on the piston, instead of the total number of strokes, the formula would apply to single-acting engines as well as double-acting, also to gas engines, &c.

In the case of gas engines, the number of strokes per minute during which explosion occurs should always be counted when indicating, as it is a variable quantity dependent on the load of the engine, even though the number of revolutions remains unchanged.

To ascertain the average pressure on the piston (P) it is necessary to have indicator diagrams : having these, the average pressure may be determined either by measurement by scales or by the use of a planimeter, which is a small machine specially devised for this purpose. In the case of steam engines, except when single-acting, it is necessary to ascertain the average pressure on each side of the piston, and then take the mean of these two pressures when calculating the indicated horse power (I.H.P.) ; but for the purpose of illustration it will only be necessary for us to deal with one diagram (fig. 48A),* as we may assume that the diagram from the other side of the piston is similar.

To ascertain the average pressure shown by this diagram, divide the diagram into ten equal parts, as shown, taking

* For fig. 48A see "*Practical Engineer* Pocket Book," 1893, page 109.

care that the dividing lines are perpendicular to the atmospheric line; then, by means of a suitably-marked scale, measure the difference between the steam and the exhaust lines at the middle of each of these divisions; add together the ten measurements thus taken, and then divide the total by 10. This will give the average distance between the steam and exhaust lines, and therefore the average effective pressure on the piston. The scale used must, of course, correspond to the spring in the indicator at the time the diagrams were taken. In the case of compound and other engines having more than one cylinder, diagrams must be taken from each cylinder if the I.H.P. is to be calculated.

Wherever a large number of diagrams has to be dealt with, or where it is necessary that the mean pressure be correctly ascertained, the measuring of the diagrams should be performed by a planimeter, as, with such a machine, the mean pressure can be more accurately and more expeditiously determined. The two principal forms of planimeter are Amsler's and the Coffin planimeter, but it is hardly necessary to enter into the manipulation of these, as full particulars are supplied with each.

The horse power thus calculated from indicator diagrams, and called the "indicated horse power," shows the total power developed in the cylinder or cylinders of the engine; but if we applied a brake to the engine in such a manner that the power given off by the engine could be accurately measured, this power would always be less than that shown by the diagrams, and would be termed the "brake" or "actual" horse power. The difference between the brake horse power (B.H.P.) and the I.H.P. is due to, and represents, the friction of the engine.

In the case of large engines it is impracticable to test the power by means of a brake; hence the only way of determining the amount lost in friction of the engine is to take friction diagrams—that is, diagrams when the engine is not driving any of its load, and such diagrams are fairly reliable as, from various experiments, it would appear that very little difference occurs in the friction of an engine, no matter what load may be on. It is wise to take diagrams for an engine alone, also for the engine and shafting, at regular intervals, say every six months, as any material increase in the friction, either of engine or gearing, would thus be detected.

Useful information can be obtained from indicator diagrams as regards the amount of steam an engine is

using, although the exact amount cannot be calculated from them, owing to the water which is almost always present in engine cylinders.

In view of the fact that a considerable portion of the water of condensation which is formed during the early period of the stroke is gradually re-evaporated during the latter portion of the stroke, the amount of steam shown by a diagram will usually be greater at the end of the expansion curve than at any other point, providing the valves and pistons are tight.

Suppose that the steam in a cylinder at the end of the expansion, and just prior to the opening of the exhaust port, is at 10 lb. pressure above the atmosphere—that is, 25 lb. absolute—and that its volume is ·95 of the total volume of the cylinder. We will further suppose that the diameter of the cylinder is such that its sectional area is 2 square feet, and that its length is 3 ft. Then the total volume of the cylinder is 6 cubic feet, and the volume of the steam at the point under consideration would be ·95 of this—that is, 5·7 cubic feet. If now we refer to saturated steam tables for the weight of a cubic foot of steam at 25 lb. pressure, and multiply this by 5·7, this will give the weight of steam in the cylinder at the point under consideration. The weight of steam, therefore, in the cylinder at the point referred to is $5·7 \times ·0625 = ·356$ of a pound.

If the valves were arranged so as to give no compression, this weight of steam would be discharged during the exhaust stroke; hence, under these conditions, it would only be necessary to multiply ·356 by the number of strokes made per hour to ascertain the weight of steam used per hour, as shown by the diagrams. Dividing this number by the indicated horse power would, of course, give the weight of steam used per hour per indicated horse power.

If, however, the valves were arranged to give a certain amount of compression in the cylinder, the whole of the ·356 lb. of steam would not be discharged, and it therefore becomes necessary to ascertain what portion would be left in the cylinder. To do this, take any point in the compression curve, and measure the pressure at this point, also the volume. For instance: Suppose that at the point taken the pressure is 5 lb. above atmosphere, and that the distance from the compression end of the diagram is ·1 or $\frac{1}{10}$ of the stroke, then the volume will be $2 \times 3 \times ·1 = ·6$ cubic foot; therefore weight of steam in cylinder $= ·6 \times ·05 = ·03$ lb.; so that instead of ·356 lb. of steam being discharged from the cylinder, $·356 - ·03 = ·326$ lb. is discharged per stroke.

Whenever convenient the point in the compression curve should be so placed that the pressure is equal to the pressure at the point taken in the expansion curve, as it will then only be necessary to refer to saturated steam tables once, and the distance between the two points will represent the period during which steam is being discharged.

Let $x = \frac{L_1}{L}$ (fig. 49),

A = sectional area of cylinder in square inches;

$\therefore \frac{A}{144}$ = sectional area of cylinder in square feet.

Let L = length of stroke in feet,

N = number of strokes per minute,

W = weight of 1 cubic foot of steam at the pressure at end of the expansion curve.

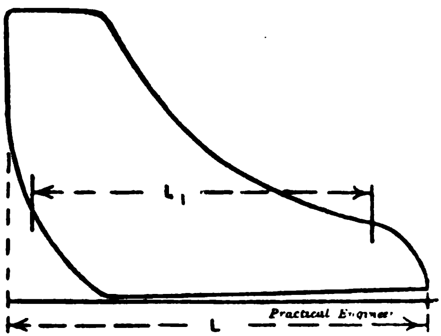


FIG. 49.—Diagram for illustrating method of calculating steam consumption.

Then $L_1 = Lx$, and volume of steam in cubic feet discharged per stroke

$$= \frac{A}{144} \times Lx,$$

and volume of steam in cubic feet discharged per hour

$$= \frac{A}{144} Lx \times N \times 60;$$

therefore weight of steam in cubic feet discharged per hour

$$= \frac{A}{144} \times Lx \times N \times 60 \times W$$

$$= \cdot416\, A L N x W,$$

and weight of steam required per I.H.P. per hour

$$= \frac{\cdot416\, A L N x W}{\text{I.H.P.}}$$

But $\text{I.H.P.} = \frac{\text{P.L.A.N}}{33000}$;

therefore weight of steam per I.H.P. per hour

$$= \frac{\cdot416\, A L N x W}{\frac{P L A N}{33000}}$$

$$= \frac{\cdot416\, x W}{\frac{P}{33000}}$$

$$= 13750\, W \times \frac{x}{P}.$$

If we call 13,750 W a constant (c), and tabulate its value for different pressures, then weight of steam per I.H.P. per hour

$$= \frac{c}{P} \times x.$$

The use of this formula, together with the following table, greatly simplifies the calculation of the water consumption without any loss in accuracy.

From the above it will be seen that the water consumption per I.H.P. can be calculated from diagrams without any knowledge as to the diameter of the cylinder, length of stroke, or speed.

In the case of compound engines the water consumption should be calculated from both H.P. and L.P. diagrams, and the larger number taken as the calculated water consumption. When looking over compound engine diagrams it is often desirable to calculate the water consumption from each pair of diagrams, as it may not infrequently happen that serious leakage of steam may be thus detected by one pair of diagrams showing a much greater weight of water than the other.

TABLE OF CONSTANTS FOR CALCULATION OF WATER CONSUMPTION.

Pressure.	13750 × W.	Pressure.	13750 × W.	Pressure.	13750 × W.	Pressure.	13750 × W.
4	154	19	664·8	34	1148·8	49	1622·3
	35·9		33		32		31
5	189·9	20	697·8	35	1180·8	50	1653·3
	34·6		33		32		31
6	224·5	21	730·8	36	1212·8	51	1684·3
	34·5		33		32		31
7	261	22	763·8	37	1244·8	52	1715·3
	34		33		32		31
8	295	23	796·8	38	1276·8	53	1746·3
	34		32		32		31
9	329	24	828·8	39	1308·8	54	1777 3
	34		32		32		31
10	363	25	860·8	40	1340·8	55	1808·3
	34		32		32		31
11	397	26	892·8	41	1372·8	56	1839·3
	34		32		31·5		31
12	431	27	924·8	42	1404·3	57	1870·3
	34		32		31·5		31
13	465	28	956·8	43	1435·8	58	1901·3
	34		32		31·5		31
14	499	29	988·8	44	1467·3	59	1932·3
	33·2		32		31		30·5
15	532·2	30	1020·8	45	1498·3	60	1962·8
	33·2		32		31		30·5
16	565·4	31	1052·8	46	1529·3	61	1993·3
	33·2		32		31		30·5
17	598·6	32	1084·8	47	1560·3		
	33·2		32		31		
18	631·8	33	1116·8	48	1591·3		
	33		32		31		

With compound engines, P, the mean pressure, must be the mean pressure for all the cylinders referred to the cylinder under consideration. Hence, if R is the cylinder ratio, and P the mean pressure in H.P. cylinder, p the mean pressure in L.P. cylinder, weight of steam per I.H.P. per hour (taking H.P. diagrams)

$$= \frac{c}{P + Rp} \times x;$$

weight of steam per I.H.P. per hour (taking L.P. diagrams)

$$= \frac{c}{p + \frac{P}{R}} \times x.$$

To determine the value x, that is $\left(\frac{L_1}{L}\right)$, possibly the handiest method, and one which does not require any special scale, is to measure by means of any of the ordinary indicator scales the lengths L_1 and L, and then divide. For instance, suppose the scale were such that L_1 measured 81 divisions, whilst L measured 92, then x would equal $\frac{81}{92} = \cdot 88$.

In the foregoing nothing has been said as regards the effect of clearance on the calculated water consumption. This has been done purposely, as, seeing that the figures calculated from diagrams are only approximate, it is rarely worth while further complicating the question by including clearance, and the effect on the figures which its inclusion would have would be very slight.

If, however, it is desired to take the clearance into account, this may easily be done by remembering that the total volume of steam in the cylinder at any point is the volume shown by the diagram, plus the clearance; hence the weight of steam should be calculated from this.

As has already been stated, when measuring a diagram for water consumption, it is usually best to select a point as near as possible to the end of the expansion curve, owing to the re-evaporation which occurs during expansion. It will, however, rarely happen that the whole of the water of condensation will have been re-evaporated even at this point; consequently the steam consumption will always be somewhat below the actual amount of steam which passes through the cylinder, and the difference will become greater as the cut-off gets earlier.

TABLE SHOWING APPROXIMATE PROPORTION OF STEAM SHOWN BY INDICATOR DIAGRAMS TO THE TOTAL WEIGHT OF STEAM USED.

SIMPLE ENGINES.

Point of cut-off.	Percentage of steam shown by diagrams.	Percentage of total amount of water present in cylinder as water at end of expansion.
Per cent. 5	58	42
10	66	34
15	71	29
20	74	26
30	78	22
40	82	18
50	86	14

COMPOUND ENGINES.

Point of cut-off.	Percentage of steam shown by diagrams.	Percentage of total amount of water present in cylinder as water at end of expansion.
Per cent. 10	74	26
15	76	24
20	78	22
30	82	18
40	85	15
50	88	12

TRIPLE-EXPANSION ENGINES.

Point of cut-off.	Percentage of steam shown by diagrams.	Percentage of total amount of water present in cylinder as water at end of expansion
Per cent. 15	78	22
20	80	20
30	84	16
40	87	13
50	90	10

With a view of assisting in making an approximate allowance for the water not shown by the diagrams, the tables on the preceding page, copied from the useful book supplied with the Tabor indicator, have been introduced.

As an instance of the use of the foregoing tables and formula, the following case may be taken, the diagrams being as shown at fig. 50.

The mean pressure for front end is 22·2, and the value of x, ·71

The mean pressure for back end is 19·6, and the value of x, ·65.

Terminal pressures being, front 11 lb., back 9 lb., above atmosphere, or 26 lb. and 24 lb. respectively absolute pressure, the water consumption becomes, therefore, front

$$\frac{892{\cdot}8}{22{\cdot}2} \times {\cdot}71 = 28{\cdot}6,$$

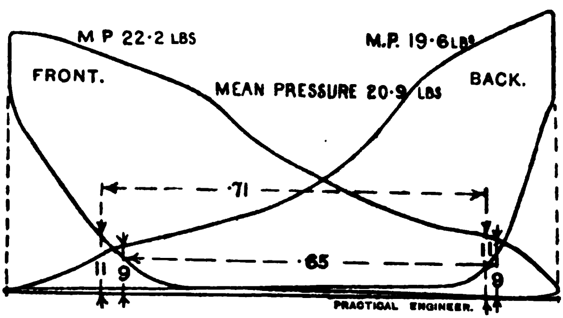


FIG. 50.—Pair of diagrams from which steam consumption is calculated.

and for the back

$$\frac{828{\cdot}8}{19{\cdot}6} \times {\cdot}65 = 27{\cdot}5\text{; average} = 28{\cdot}05.$$

Now, seeing that the point of cut-off occurs at about ·3—*i.e.*, 30 per cent—the above quantity will represent 78 per cent of actual water consumption, which will therefore be about 36 lb. per horse power per hour.

The weight, calculated from the diagrams in the manner just explained, will, of course, be only approximate, as it is impossible to say definitely what proportion is present in the cylinder as water, as this varies with the size of the cylinder,

the arrangement of the cylinder, whether covered by non-conducting material, &c. The only strictly reliable way is to make careful measurements of the water evaporated in the boiler, or where possible, as in the case of surface-condensing engines, by measuring the weight of water passed from the condenser.

By calculating the water consumption from indicator diagrams taken under different conditions of working, very valuable information may frequently be obtained. For instance, it is possible to determine approximately the most economical load to put on any engine, and there can be no doubt that in a large number of cases material improvement might be obtained if the power required were more suitably proportioned to the engine; and in connection with all new

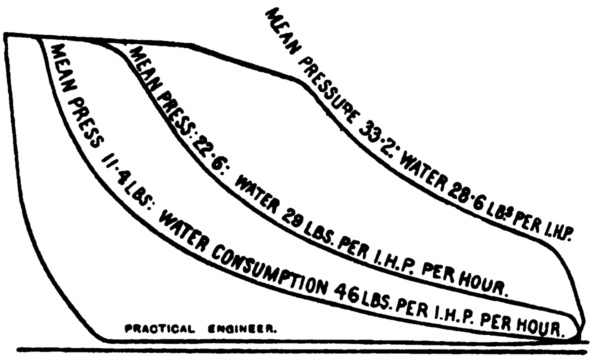


FIG. 51.—Diagrams showing reduced economy as cut-off gets earlier owing to increased cylinder condensation.

engines, information respecting the best load is useful when deciding on the question of the best size of engine.

In many text-books on the steam engine the fact that increased work can be obtained from a given quantity of steam by utilising its expansive property is fully gone into; but, unfortunately, this statement is not unfrequently given without any qualification; consequently many people believe that the earlier the cut-off in any cylinder, the greater the economy. This, however, is only true within certain limits, as it is found that after a certain point in any engine the loss introduced by cylinder condensation more than counterbalances the gain by increased expansion. The rapid rate at which condensation increases as the cut-off gets earlier

is shown by the table already given. At fig. 51 three diagrams, taken from a horizontal non-condensing engine, are shown, and the calculated water consumption is given on each curve. From these it will be seen that instead of decreasing as the cut-off got earlier, the water consumption increased, proving that, although the steam was expanded to a greater extent, the gain thus obtained was outweighed by the increased condensation losses.

A further illustration of this point is shown at fig. 52.

When considering the diagrams just given, it must also be borne in mind that the water calculated from the early cut-off diagrams does not in all probability represent such a high proportion of the whole as is the case with the later

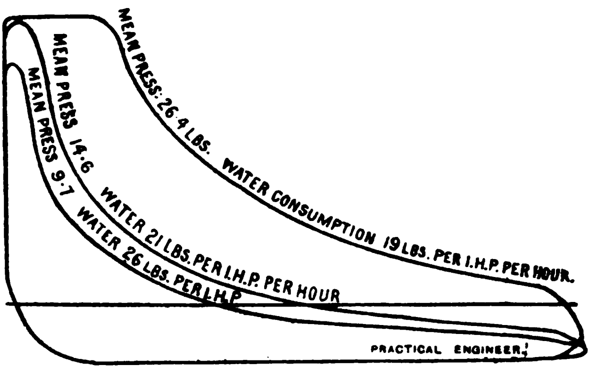


FIG. 52.—Diagrams from condensing engine, illustrating same point as fig. 51.

cut-off diagrams; and to make the comparison more complete, a portion of the power shown by each diagram should be deducted, so as to allow for engine friction, and thus give the water for each effective horse power; and it should further be remembered that to develop any given power a much larger engine, and consequently more expensive, would be required with the early cut-off than with the later cut-off.

Taking all the above-mentioned points into account, it is evident that, so far as economy is concerned, it is better to err on the side of heavy loading rather than light loading, and the not uncommon practice of putting down an engine much too large for the work, because there is a probability that at some future date there may be a little increase of load, should be avoided.

Compound engines are, as a rule, more economical than single-cylinder engines, probably mainly owing to the decreased range of temperature in each cylinder causing the losses by cylinder condensation to be reduced. This may be illustrated by a diagram (see fig. 53).

The curve A B represents the usual expansion curve for a single-cylinder engine. The reason for the great difference between this and the hyperbolic curve has already been explained. If now we take the case of a compound engine, with a cylinder ratio of 3 to 1, and suppose that the terminal pressure in the low-pressure cylinder is the same as in the former case, then, as the range of expansion in the low-pressure cylinder is less than in the simple engine, the

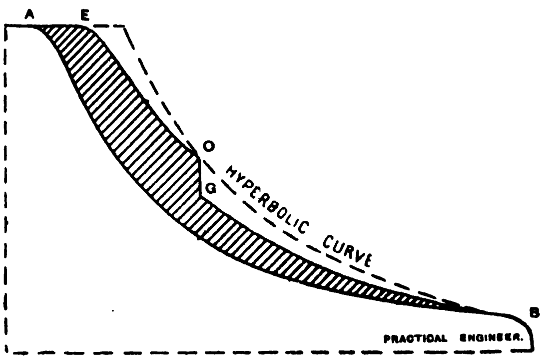


FIG. 53.—Diagrams illustrating reason for compound engine being more economical than single engine.

low-pressure expansion curve will more nearly approach the hyperbolic, as shown by the line B G; then the vertical line G O will represent the loss of pressure between the cylinders and the curve O E, the expansion curve for the high-pressure cylinder.

The horizontal length of the high-pressure diagram is one-third the length of the low-pressure, 3 being the cylinder ratio.

The shaded area A B G O E represents the gain due to the compound system.

Although the compounding of engines usually improves economical working, it introduces additional sources of loss, such as drop of pressure between the cylinders, increased

surfaces for radiation, increased throttling in pipes and ports, increased friction, &c. If it is desired to obtain some information as regards the extent of the former of these, it is wise to combine the diagrams from the two cylinders.

To do this, draw the line A B (fig. 54) so that the distance between A and B represents to any known scale the capacity of the low-pressure cylinder. Then make A X so that it represents to the same scale the clearance in the low-pressure cylinder; then erect the vertical lines X Y, A C, and B D.

From X mark X E so that the distance between X and E represents the clearance in the high-pressure cylinder. Then make E F to represent the volume of the high-pressure

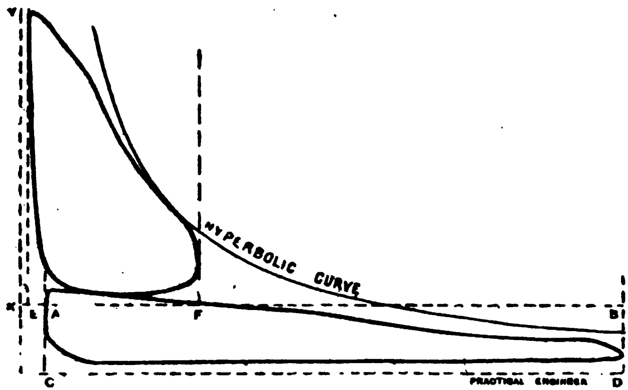


FIG. 54.—Combined diagrams from compound engine.

cylinder to the same scale. The diagram taken from the high-pressure cylinder should then be shortened so as to fit between the verticals drawn from E and F, and, similarly, the low-pressure diagram should be placed between the verticals A C, B D, the same scale of pressures being maintained throughout both diagrams. The hyperbolic curve should then be drawn so as to just touch either of the diagrams.

The space between the two diagrams represents the loss between the cylinders.

In the instance just taken the diagrams which have been combined are shown at fig. 55. To change the length of either diagram so as to suit the combined arrangement, probably the simplest method is to divide each length (A B

and E F), also each diagram, into ten divisions, then measure the pressure at each of the divisions on the ordinary diagram, and transfer these pressures to the corresponding lines on the combined diagram.

If either cylinder were receiving much more steam than the other one, a combined diagram would show this at once. An instance of this is given at fig. 56, which shows the combination of diagrams taken from a compound condensing

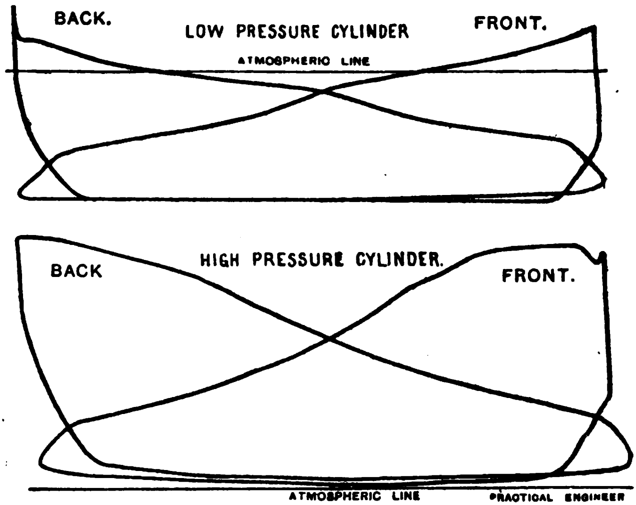


FIG. 55.—Diagrams combined at fig. 54.

engine. The greater weight of steam shown by the low-pressure diagrams being due to steam entering the low-pressure cylinder from an auxiliary valve connected to the low-pressure cylinder to facilitate starting, of course it is essential that the diagrams used for combining be taken under similar conditions of load and pressure, otherwise misleading deductions may be made; in fact, all diagrams from a compound engine should be taken with the load on the engine and the boiler pressure as steady as possible.

CHAPTER IV.

STEAM-ENGINE valves, and the various methods of dealing with them, have been gone into very exhaustively by numerous writers, but if the general subject under consideration—viz., indicator diagrams—is to be dealt with fairly and fully, it seems desirable that some mention at least be made of such valves, and the method of controlling them. The reader is, however, referred to one or other of the books more especially devoted to this branch, if he desires further information.

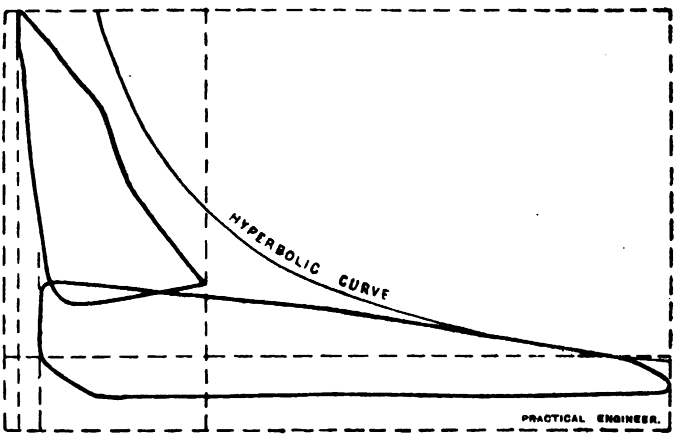


FIG. 56.—Combined diagrams showing leakage of steam into low-pressure cylinder.

The distribution of steam in a cylinder—that is, the regulation of the points of admission, cut-off, &c.—may be, and is, performed by a great variety of valves and valve gears; but by far the commonest is that known as the simple slide valve. As the fundamental principles relating to the simple slide valve apply in a great measure to most other valves, we will commence by a brief consideration of them.

The simple slide valve consists of a rectangular box, of the form shown in fig. 57. One of the faces (F) is planed, and made to move truly over the surface (G), out of which the steam ports open. From the drawing it will be seen that two of these passages, A and B, lead into opposite ends of the cylinder, while the other (C) leads away from the cylinder.

The two former are termed the steam ports, whilst the third is called the exhaust port.

In its simplest form the valve is designed so that, when in the middle of its range of movement or travel, the two steam

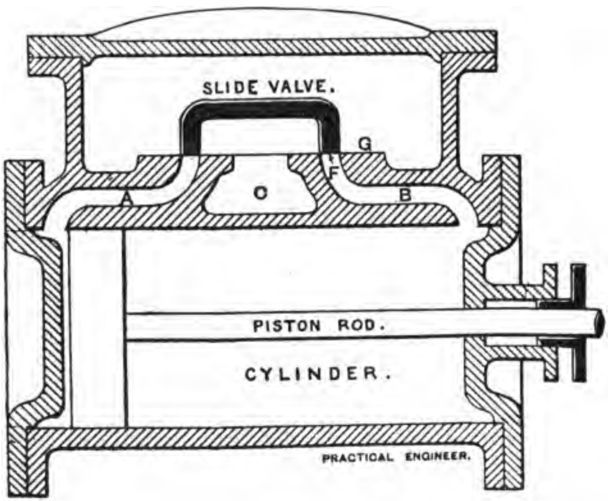


FIG. 57.—Longitudinal section through cylinder and steam chest.

ports are just, and only just, covered by the valve; hence, as long as the valve remains in this position, steam cannot either enter or leave the cylinder. If, now, the valve were moved a little to the right, steam would flow down the steam port on the left side to the cylinder, whilst at the same time the steam on the right side of the piston would flow into the cavity in the valve, and from there down the exhaust port (C) to the atmosphere or condenser, according as the engine is non-condensing or condensing. Similarly, if the valve were moved to the left, steam would flow into the right-hand

end of the cylinder, while the waste or exhaust steam at the left-hand side of the piston passed away out of the cylinder.

By suitable driving arrangements, such a valve might easily be arranged to admit steam to either end of the cylinder just as the piston reached the end of the stroke; it would thus press on one side of the piston, and at the same time the valve would allow the steam to escape from the opposite end of the cylinder, and thus relieve the pressure on that side of piston, with the result that the piston would be pressed forward until the flow of steam was reversed by the valve.

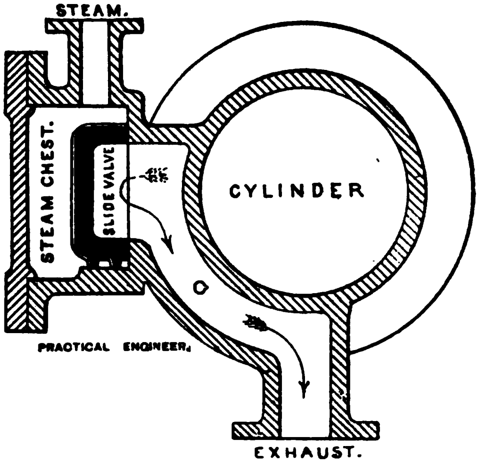


Fig. 57.—Section through middle of cylinder showing exhaust passage.

Ordinary slide valves, and, in fact, almost all types of valves, usually receive their motion from eccentrics secured to the crank shafts.

As the eccentric is keyed or otherwise secured to the crank shaft, it revolves with it; hence its centre describes a circle round the centre of the crank shaft, and, consequently, the motion derived from an eccentric is exactly the same as would be derived from a small crank, and the travel of the eccentric rod would be equal to the diameter of the above-mentioned circle—that is, twice the distance between the centre of the crank shaft and the centre of the eccentric block.

Let A B (fig. 58) represent the path of an engine piston, and C D E be the circular path of the crank pin, and the distance A C equal the length of the connecting rod. If the crank pin were at E, the position of the piston might be found by setting compasses or trammels to the length of the connecting rod, and drawing an arc cutting A B at F; then F is the required position of the piston. To represent this position on C B, set the trammel or compass centre at F, and describe the arc E G; then C G would equal A F, and would denote the distance the piston had travelled from one end of its stroke when the crank pin was at E. If the connecting rod were lengthened relatively to the radius of the crank, the curve E G would become flatter, and the point G would be nearer H, which is obtained by dropping a perpendicular from E. Hence, if the slight effect of the connecting rod be neglected, the position of the piston for any point of the crank pin may be determined by simply dropping a

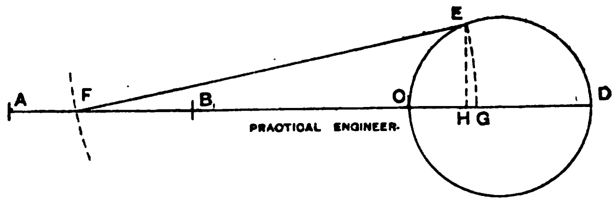


FIG. 58.—Sketch showing obliquity of connecting rod.

perpendicular from the point in question. This method would also be applicable for finding the position of the valve for any position of the eccentric.

For our present purpose it will be sufficiently accurate if the length of both the connecting rods and the eccentric rods be neglected, and this will considerably simplify the problems to be dealt with.

In the valve just considered, the faces were just of sufficient width to cover the ports A and C, without overlapping, when the valve was in its mid-position. Valves usually, however, overlap the ports, especially on the steam side. This overlap, when on the steam side of the valve, is called "steam lap"; when on the exhaust side of the valve it is called "exhaust lap." In fig. 59 steam lap is shown by diagonal shade lines, while exhaust lap is shown by solid black. It must be remembered that the lap is always measured when the valve is in the middle of its travel or range of movement.

There is one other term in connection with valves to be noted, namely, "lead." This term is used to express the amount the valve is open for steam when the piston is at the end of its stroke; thus, $\frac{1}{4}$ in. lead would mean that the steam

FIG. 59.—Sketch showing steam lap and exhaust lap.

port was $\frac{1}{4}$ in. open when the piston was just at the end of its stroke.

When a valve has no lap or lead, the eccentric would be placed at right angles to the crank, so as to cause the valve to be at mid-travel—that is, just on the point of opening for

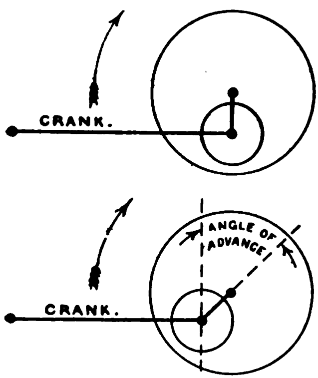


FIG. 60.

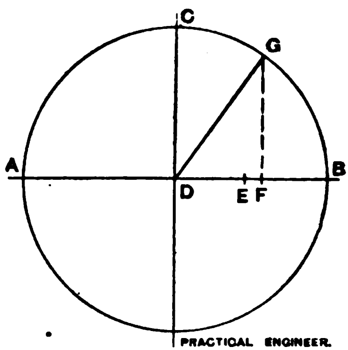


FIG. 61.

steam and exhaust when the piston is at the end of the stroke. If, however, the valve has lap, then the eccentric will require to be moved beyond its central position by an amount equal to the lap for it to be just on the point of opening for steam; consequently the angle between the

centre line of the eccentric and the centre line of the crank will be greater than a right angle, and this increase is usually termed the "angle of advance." The above-mentioned positions of eccentric and crank are illustrated at fig. 60. If lead is required, the eccentric would require advancing a little further.

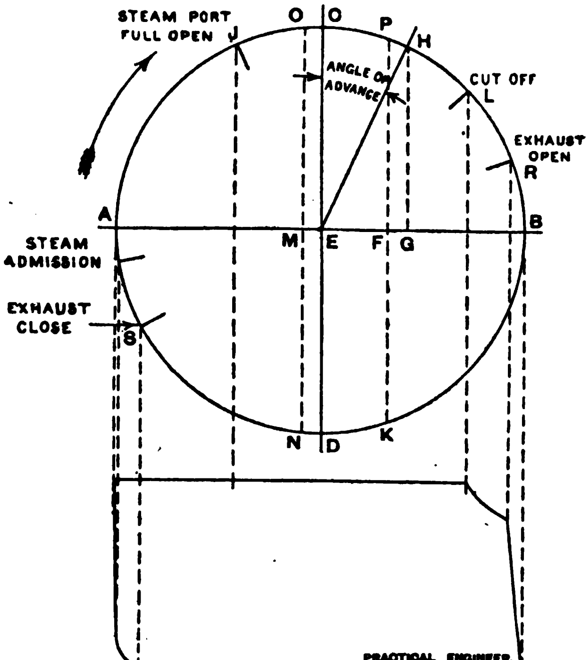


FIG. 62.—Method of determining points of cut-off, &c., for a slide valve.

Referring to fig. 60, it will be seen that, as the crank is on the left-hand centre, the direction of the valve's movement must be such as to cause the left-hand steam port to be opened; consequently the valve must move to the right, and hence the shaft must rotate in the direction shown by the arrow. From this it will be seen that the direction in which an engine would turn may be readily observed from the position of the eccentric relative to the crank.

To determine the angle of advance required for any given lap and lead, it is only necessary to draw a circle A B C, fig. 61, having a diameter equal to the travel of the valve; then from the centre D draw D E equal to the lap and E F equal to the lead; then from F erect a perpendicular F G, to cut the circle at G. The angle C D G will be the angle of advance. The reason for this is as follows: The perpendicular G F, where it cuts A B, gives the position of the valve when the centre of the eccentric is at G, as has been already shown. Therefore, when the eccentric is at G the valve is at a distance D F from its central position; but as the lap on the valve is D E, the valve must move this amount from its central position before it commences to open, after which any further movement causes opening of the steam port. Therefore the remaining movement E F is the amount the port would be open when the eccentric is at G, or E F equals the lead, if the eccentric angle of advance is C D G, and, *vice versa*, C D G represents the requisite angle of advance to give lead equal to E F with lap equal to D E.

Having seen how to determine the angle of advance for the eccentric for any given amount of lap or lead, and having given the travel of the valve, it becomes easy to determine the points of cut-off, exhaust opening and closing, and this may be done as follows: Draw circle A C B D, fig. 62, having its diameter equal to the travel of the valve; then draw the vertical and horizontal centre lines A B and C D, crossing the centre at E.

Make E F equal to lap on the steam edges, and F G equal to the lead. Draw vertical line G H, then angle C E H equals angle of advance, and angle A E H equals angle between crank and eccentric, and this is a fixed angle for all positions of the crank; hence it is only necessary to determine the position of the eccentric when cutting off, &c., and measure backwards on the circle an angle equal to A E H, to determine the corresponding position of the crank.

Starting with the crank at A, and supposing that it is moved round in direction shown by arrow. When the eccentric is at B the valve will be at one end of its travel; hence the port opening will be at its maximum. The crank will then be at J, this position being simply obtained by measuring back with compasses, so that B J equals H A. When the eccentric is at K, obtained by dropping a perpendicular from F, the valve will be a distance of E F from its

central position—that is, a distance equal to the steam lap; hence the valve will just have closed the steam port, so that by measuring back as before we get point L as the position of the crank at cut-off.

If the valve has no exhaust lap, the exhaust will commence to open when the valve reaches its central position, but if it

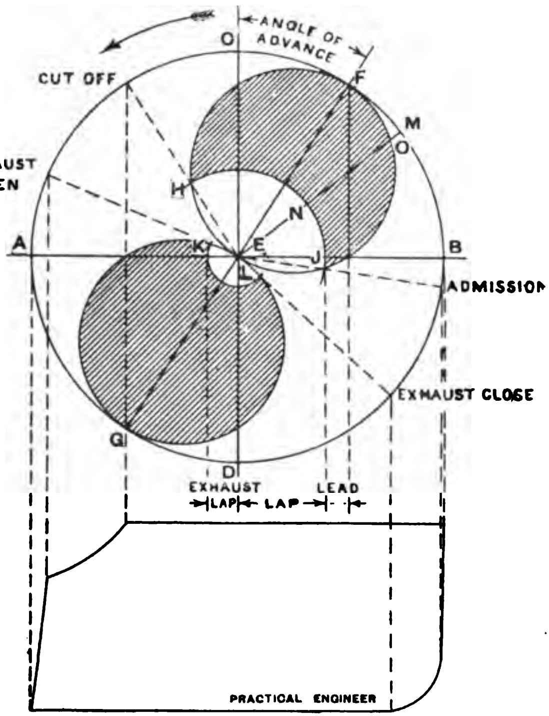


FIG. 63.—Zeuner's valve diagram.

has lap, say, equal to E M, then the exhaust will open when the eccentric is at N, and will close when it is at O, the corresponding position of the crank being obtained as before. The admission of steam to the cylinder would commence when the eccentric reached P. By dropping

perpendiculars from each of crank positions thus obtained, the corresponding positions of the piston would be given; consequently an approximate outline of the indicator diagram might then be drawn.

A handier method of determining the points of cut-off is that known as Zeuner's. The disadvantage of this, and the

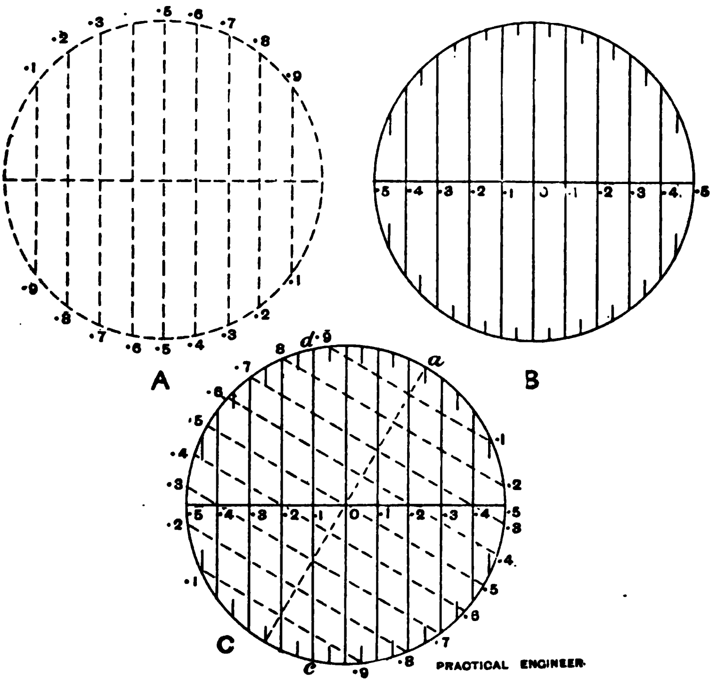


FIG. 64.—Handy method of determining points of cut-off, &c., for a slide valve.

several other forms of valve diagram, so far as a student is concerned, is that the reason for the various steps is not so obvious. This, however, is not of much importance, as only a little study is required to thoroughly understand the methods adopted.

Determination of points of cut-off, &c., by Zeuner's method, having given the travel of valve, laps, and lead.

Draw circle A B C D, fig. 63, of diameter equal to travel of valve, as before, then from centre mark lap and lead, and determine angle of advance as before. Let C E F be the angle of advance, produce F E to G, and draw the two circles F H E J and K G L E, the diameters being half the travel of the valve. These circles are called the valve circles. Then draw arcs of circles H J and K L, with radius equal to steam and exhaust lap respectively, and cutting the valve circle in H, J, K, and L. From E draw radial lines through the points of intersection of the valve circles with the lap circles. These lines will be the positions of the crank at the points of cut-off, &c. Radial lines drawn through the shaded portions of the valve circles will give the port opening for the various positions of the crank; thus when the crank is at M the port opening will be N O.

For information as to the various ways of utilising Zeuner's valve diagram for the solution of problems in connection with steam-engine valves, Zeuner's or some other treatise devoted to valves should be consulted.

Where many valves have to be considered, the following simple arrangement will be found very convenient, and if carefully used will give very accurate results, especially as the effect of the connecting rod can be easily introduced.

Take a piece of fairly stiff drawing paper and a similar piece of tracing paper, and on each draw a circle of any diameter, but preferably 10 in., and divide the centre line of each of these circles into ten equal parts; then draw perpendicular lines from each of these parts. For the convenience of illustration, these circles have been shown at fig. 64, one with full lines (B), the other with dotted lines (A). If the circles are drawn to a large scale, say 10 in. diameter, each of the horizontal divisions may be further divided into ten parts. In the illustration they have been subdivided into two parts.

As an example of the use of these circles, we will take the case of a valve having a travel of 5 in., steam lap 1 in., exhaust lap $\frac{1}{2}$ in., lead $\frac{1}{4}$ in. Then the ratio of the steam lap plus lead to the travel of valve, would equal $\frac{1\frac{1}{4}}{5} = \cdot 25$, or $2\frac{1}{2}$ tenths, while the ratio of the exhaust lap to the travel would be $\frac{\frac{1}{2}}{5} = \cdot 1$ or $\frac{1}{10}$, and the steam lap itself is 1 fifth or 2 tenths of the travel. Place the diagram on tracing paper—that is, the one drawn in dotted lines (A)—over the other one (B), as shown at C, in such a manner that the centre line

terminates at the vertical line drawn from ·25—that is, mid-way between ·2 and ·3, ·25 being the ratio of lap plus lead to the travel. From previous explanation, it will be seen that the angle between the vertical centre line and the dotted centre line is the angle of advance. As the lap equals ·2 of the travel, the point at which the vertical dropped from ·2 cuts the circle will be the position of the eccentric at cut-off, and the angle represented by the arc or portion of the circle *a b* will be the angle moved through by the eccentric, also by the crank from the commencement of the piston's stroke to the point of cut-off. By placing the dotted circle as shown and described, this angle is readily measured in terms of the stroke, and in this case is ·8.

If there were no exhaust lap on the valve, the exhaust opening would commence when the valve reached its mid-

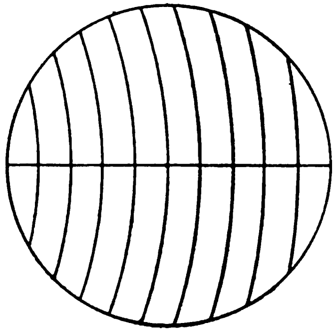


FIG. 65.—Diagram showing mode of dividing circle A, fig. 63, when connecting rod is taken as being four times the radius of crank.

position, which, as will be seen, would be at about ·93 of the stroke; but as there is exhaust lap equal to 1 tenth of the travel, the exhaust opening would occur at point *c*—that is, 1 tenth division to the left of the centre, and this equals about ·98 of the stroke. The exhaust would close for similar reasons at *d*—that is, about ·87 of the stroke.

If it is desired to take into account the length of the connecting rod, all that is necessary is to have the dividing lines on the superposed circle drawn to a curve, the radius of which is equal to the length of the connecting rod, as at fig. 65. Generally, connecting rods are made from four to six times the radius of the crank, so that if three circles are drawn, one with curves having a radius equal to four times

the diameter of the circle, another five times, and the remaining one with six times, these will cover all ordinary cases.

It is frequently handy to be able to determine the amount of lap which a valve should have to give a certain point of cut-off, or to know the point of cut-off which a certain

TABLE OF LAPS, &c.

Point of cut-off.	Ratio of lap to travel of valve.	Point of stroke at which exhaust would open and also close.
$\frac{1}{4}$	·43	·75
$\frac{3}{10}$	·42	·78
$\frac{3}{8}$	·4	·81
$\frac{4}{10}$	·39	·82
$\frac{1}{2}$	·35	·86
$\frac{6}{10}$	32	·89
$\frac{5}{8}$	·31	·9
$\frac{7}{10}$	·28	·92
$\frac{3}{4}$	·25	·94
$\frac{8}{10}$	·23	·95
$\frac{7}{8}$	·18	·97
$\frac{9}{10}$	·16	·98

amount of lap would give, and at the same time have some idea as to the points of exhaust opening and closing, without adopting any geometrical methods.

In such cases, the foregoing table will be found useful. In compiling it, the lap has been taken in fractions of the valve travel, and it has been assumed that the valves have no lap on the exhaust edges.

CHAPTER V.

Defective Indicator Diagrams.

If a considerable number of diagrams taken from engines working under ordinary conditions were collected together, a large number of these would, almost certainly, be found defective in some way; that is to say, they would show that the engines were not working under the most favourable conditions, or they in themselves would be inaccurate, and would give a misleading idea as to what was going on in the cylinder of the engine.

The number of ways in which diagrams may be defective is almost unlimited, but may be divided into two principal divisions, of which the first would include defects brought about by the valve gear or other working parts of the engine not being in order or in proper adjustment.

The second division would include those diagrams which give a false representation of the action or pressures of the steam in the cylinder, either through the indicator or its connections being out of order, through the indicator being improperly handled, or through the arrangement for driving the indicator barrel being incorrect in some way.

The first-mentioned class is a very large one, including as it does defects brought about by incorrect setting of the valves or eccentrics, faulty design of valves, ports, or other passages leading to or from the cylinder, leakage of steam past the pistons or valves, unsuitable air-pump arrangements, deficiency of the water supply for condensing purposes, unsuitable load, &c.

Taking the first mentioned of these, we will commence with the eccentric, and see in what way the position of this affects diagrams which would be obtained by the indicator.

From previous explanations readers will, no doubt, quite understand that every operation performed by a simple slide valve is timed by the eccentric; consequently any change in the position of the eccentric alters the distribution of steam in the cylinder at every point. Hence, if the eccentric is advanced—that is, if the angle between the centre line of the eccentric and that of the crank is increased—the points of admission, cut-off, release, and compression will occur at earlier periods of the stroke, and this will be equally true for both sides of the piston. Such diagrams are illustrated at fig. 65, where the thin line shows the

diagram taken when all is properly set, while the thicker line shows the effect of having the eccentric set too far forward. From these it will be readily seen that each of the above-mentioned points is earlier by the advancement of the eccentric.

If, on the other hand, the eccentric be behind its proper position, all parts of the steam distribution will be late, and this is a much more important defect than the one previously described, as a late admission of steam to a cylinder, apart from considerations of economy, is frequently very prejudicial to the smooth running of the engine, especially if the latter works at a high speed. The deficient compression and late exhaust opening, which it generally also brings about, are objectionable on the same grounds. The

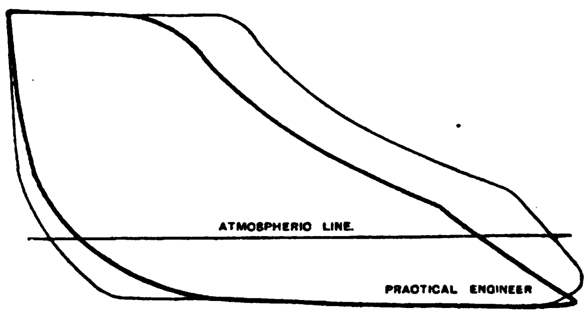


FIG. 65.—Diagram illustrating a too great advance of eccentric.

reason for this has already been fully explained in connection with theoretical diagrams.

At fig. 66 three diagrams are shown illustrating different degrees of lateness of the eccentric.

If the ports and valves of the engine are well designed, and there is no undue throttling in the pipes or regulating valves, the pressure of steam in the cylinder should, as has already been explained, be fairly close to the boiler pressure; but if the eccentric is late, the lateness of admission which this will cause will prevent the free access of steam to the cylinder during the early portion of the stroke. In many instances, the only way in which diagrams will show lateness is by a slight rounding at the admission portion, as shown at (*a*), fig. 66, while in others the lateness will be more pronounced, as at (*b*) and (*c*).

In the case of a simple slide valve, lateness of the eccentric, in addition to causing late admission, also causes lateness of all other points. Where the exhaust and steam valves are

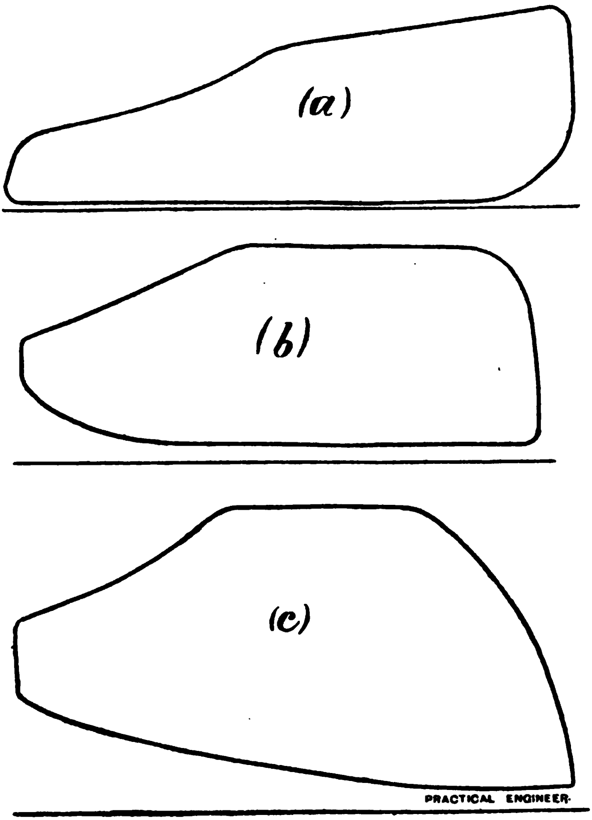


FIG. 66.—Diagrams illustrating late eccentric.

separate, and are actuated by different eccentrics, lateness of the steam eccentric would not in any way affect the exhaust valves, and *vice versa*. Thus the exhaust opening and closing might be early and still the admission be late, as at

fig. 66*a*. In this case it will be noted that a loop is formed at the compression end of the diagram; this is due to the lateness and to the fall of pressure by condensation or leakage at the end of the stroke.

We will next take the case of an engine having the lap on the steam edges of the valve very unequal—say lap on front steam edge greater than lap on back steam edge of valve.

From previous matter we know that if the lap on a valve

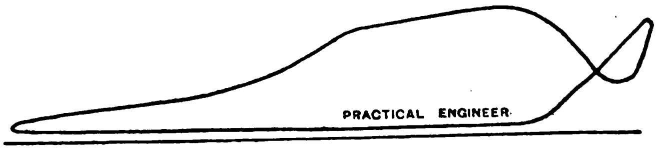


FIG. 66A.—Diagram illustrating late admission.

is increased it becomes necessary for the angle of the eccentric's advance to be increased. Hence, if the eccentric had been set to suit the smaller lap at the back end, it would be too late for that at the front end; consequently the admission at the front end would be later than at the opposite end. The point of cut-off would, however, be a little earlier, as the extra lap would cause the steam port to be closed rather sooner in the stroke. This defect of unequal lap on the

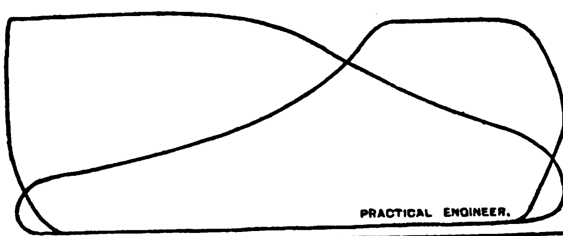


FIG. 67.—Diagram illustrating unequal lap on steam edges.

steam edges does not in any way affect the points at which the exhaust port is opened and closed; consequently the only alterations in the form of the diagrams are those due to the late admission and the earlier cut-off at the front end. An illustration of this defect is given at fig. 67.

If, instead of the inequality being at the steam edges of the valve, it were at the exhaust edges, the points of admission and cut-off would not be affected, but the exhaust

at the end having the most lap would open later and close earlier than at the opposite side of the piston; that is to say, if the lap on the front exhaust edge of the valve were greater than on the back edge, the exhaust opening at the front end would be later than at the back, whilst the compression would occur earlier. For diagrams showing this defect see fig. 68.

We have now seen the effect of unequal laps on the valve itself. This is an uncommon defect, but similar inequality, owing to the setting of the valve, is very common indeed, and is brought about either by the valve not being in its proper position on the spindle, or to the eccentric rod being of the wrong length. In this case, however, both the exhaust and the steam points are altered.

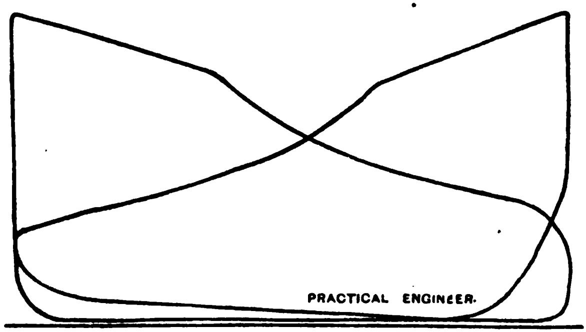


FIG. 68.—Diagram illustrating unequal lap on exhaust edges.

Take first the case in which the valve is too near the back end. Under these circumstances the lap on back steam edge will be above its intended quantity, and greater than on the front steam edge; whilst the lap at the back exhaust edge will be diminished, and that at the front exhaust edge increased.

The increased steam lap on the back end will, as has been explained in a previous case, cause later admission and earlier cut-off at that end, whilst at the same time the decreased lap at the front end will reverse the case at the front. The alterations in the exhaust lap will cause earlier exhaust opening and later closing at the back end, owing to the exhaust lap at that end being decreased. At front end the exhaust port will be opened later and closed earlier.

As a rule the slide valves of stationary engines are fixed on their spindles by a pair of nuts at each end. With this arrangement it is only necessary to move the valve a little towards the front end by means of the nuts to remedy the irregular setting.

If the valve were too near the front end, the steam distribution at that end would correspond to that at the

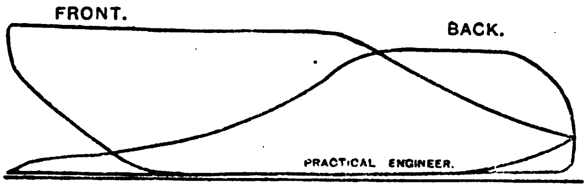


Fig. 69.—Diagram taken with valve unequally set.

back end in the former case—that is, the admission would be later, the cut-off earlier, the exhaust opening earlier, and the exhaust closing later; and these conditions would be reversed at the back end. Tabulating these changes, we get the following:—

Condition of valve setting.	Admission.		Cut-off.		Exhaust opening.		Compression begins	
	Back.	Front.	Back.	Front.	Back.	Front.	Back.	Front.
Valve too near back	Late	Early	Early	Late	Early	Late	Late	Early
Valve too near front	Early	Late	Late	Early	Late	Early	Early	Late

For an instance of unequal valve setting see fig. 69.

When such defective valve setting is discovered in any engine steps should be taken to remedy it, as it is prejudicial to economicial working, as well as tending to cause irregular turning. It has already been explained that with a simple slide valve the points in the distribution of the steam are very dependent on one other—that is to say, for instance, the point of cut-off cannot be altered without affecting one or more of the other points, and as the cut-off gets earlier, the points of exhaust opening and closing will get earlier. Further, it has been explained that, speaking generally, a slide valve should not be arranged to cut off earlier than at four-tenths or one-half of the stroke.

CHAPTER VI.

WHERE it is desired for economical reasons to cut off the supply of steam to the cylinder at an early period of the piston's stroke, it is usual to adopt some form of compound valve—that is, one valve working on the top of the other; or instead of this some arrangement in which separate valves are used to regulate the steam supply and the exhaust.

A very common form of the first-mentioned arrangement is that known as Meyer's, which consists of a flat valve A, fig. 70, having its back face planed as well as the front face. Two flat plates work on the back of this valve, and the spindle connecting them has right and left handed screws

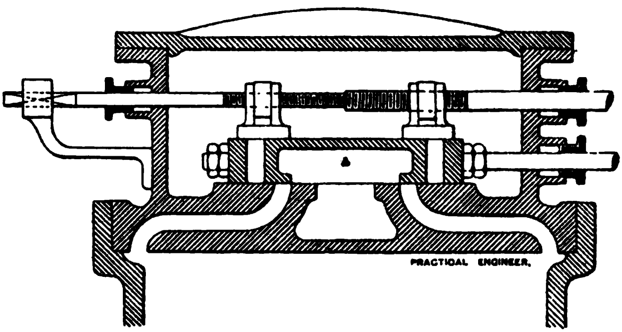


FIG. 70.—Meyer's Valve.

on it, so that the distance between the plates may be regulated for the purpose of altering the cut-off.

In designing such a valve the main valve should be arranged specially with a view of getting suitable compression and exhaust opening; after this the expansion plates should be proportioned to give the requisite range of cut-off.

A very prevalent fault with this type of valve gear is that the main valve has too little lap on the steam edges. This results in there being very little compression and in the exhaust opening late in the stroke (see fig. 71). The deficiency of steam lap in some cases leads to a defect peculiar to compound valve arrangements, namely, re-admission of steam to the cylinder after the proper point of

cut-off, although it may also occur when there is lap on the main valve. (See fig. 72.) This defect is entirely due to the valves not having been properly designed, but there is much less risk of it occurring, and it is easier to design the valves, when the main valve is given a fair amount of lap on the steam edges, and cuts off, say, at about half or six-tenths of the stroke.

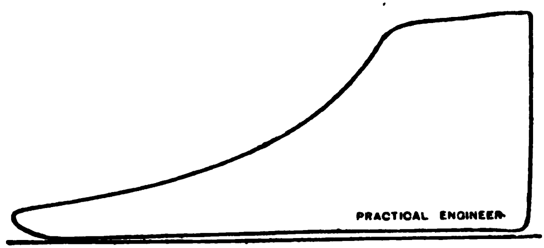


FIG. 71.—Diagram illustrating deficient compression and late exhaust opening with compound valve arrangement.

A similar defect is sometimes found when the expansion valve is of the grid type, owing to bad design. Fig. 73 shows a very bad case of this. The remedying of this defect would certainly cause very material reduction in the amount of steam used by this engine.

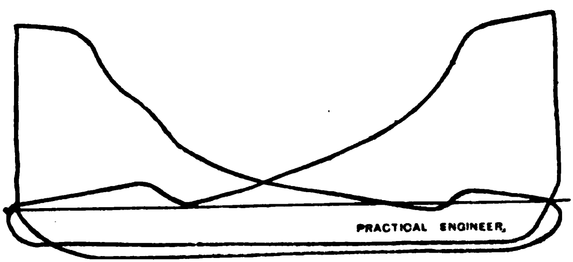


FIG. 72.—Diagrams illustrating re-admission of steam to cylinder owing to re-opening of expansion valve of Meyer's type.

With expansion valves on non-condensing engines it frequently happens that the expansion curve passes below the atmospheric line, with the result that when the exhaust port is opened the pressure in the cylinder rises instead of falls, and a loop is thus formed at the toe of the diagram, as

shown at fig. 74. In determining the mean pressure of a diagram of this kind, the pressures measured below the atmospheric line must be added together and subtracted from the sum of those measured above, and the result must then be divided by the total number of divisions taken, usually ten, to give the mean pressure.

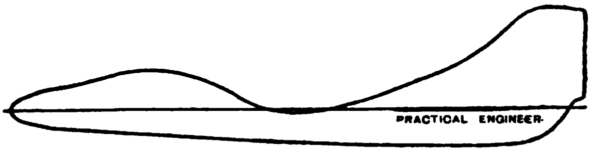


FIG. 73.—Diagram illustrating re-admission of steam through re-opening of grid expansion valve.

An extreme case of a defect of this character is shown at fig. 75. This pair of diagrams shows that instead of the steam doing useful work, an amount of work was expended on it in taking it through the cylinder, this work, in addition to the friction, being obtained from another cylinder connected to the same crank shaft. The high back

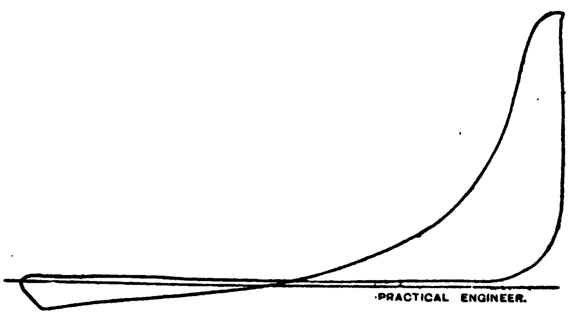


FIG. 74.—Diagram taken from non-condensing engine with expansion below atmosphere.

pressure was due to the exhaust steam being utilised for heating purposes.

Where steam pipes are of great length, and exposed to the cooling influences of the atmosphere, there will be considerable loss of pressure between the boiler and engine, owing to condensation of steam, and the water of condensation would be likely to cause trouble in the cylinder unless

very good draining arrangements were adopted. In view of this it is very desirable to protect all steam pipes externally by some non-conducting material, and this is especially so if the pipes are long or are exposed to the outside atmosphere. Again, all sharp bends or abrupt changes of sectional area should be avoided, and as far as possible the openings in all valves should be at least equal in area to the sectional area of the steam pipe.

If indicator diagrams from a cylinder show that the initial pressure is below that which might be expected, indicator taps should be fixed (*a*) on the steam chest, (*b*) on the steam pipes just on the boiler side of the starting valve, (*c*) on the boiler itself; and the indicator should be applied to each of these taps.

These diagrams would show the pressures at the above-mentioned points, and it would then be easy to note what drop of pressure occurred between (*a*) the boiler and the starting valve, (*b*) the starting valve and the steam chest,

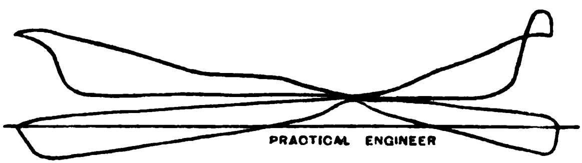


FIG. 75.—Diagram showing expansion below atmosphere, and high back pressure

(*c*) the steam chest and the cylinder. By this means the part causing throttling would be located, and proper steps could then be taken to remedy the defect. If in the steam pipes, these would require altering to do away with sharp bends, or would require substituting by pipes of larger diameter; if in passing through the starting valve, and this was full open when diagrams were taken, this valve would in all probability be found on examination to be badly designed or in some way deficient in area of opening.

When taking diagrams from a steam chest it is essential the barrel of the indicator be actuated exactly as it would be when indicating the cylinder—*i.e.*, so as to move coincidently with the piston.

An illustration of a steam-chest diagram is given at fig. 76. This diagram is such as would usually be obtained from the steam chest of an average engine fitted with an automatic expansion valve gear. The explanation of the diagram is as follows: At point A the steam port leading

to the front end of the cylinder is opened, causing a sudden drop of pressure in the steam chest, and consequently a corresponding fall of the indicator pencil, thus marking the portion A B. Then during the admission period the pressure usually falls slightly, as shown by B C. At point C the cut-off occurs, with the result that the flow of steam is suddenly checked, causing a rapid rise of pressure C D in the steam chest. Not infrequently the momentum of the steam rushing towards the steam chest is sufficient to cause the pressure in the chest to rise above the boiler pressure, as is shown on the diagram; the boiler pressure is shown by the dotted line. After the point of maximum pressure is reached the pressure usually falls slightly, as shown,

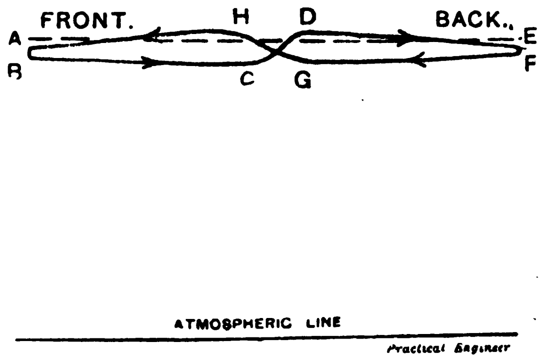


FIG. 76.—Steam chest diagram from ordinary automatic cut-off engine.

until the steam port at the opposite end is opened, after which the cycle of operations is repeated. The direction in which the pencil moves is shown by arrow points on the diagram.

If the pipes are well proportioned and arranged, and there is no throttling at the valves fixed to the pipes, the lower lines of the diagram would be close to the upper lines, whereas if throttling occurred at any part the lines B C and and F G would slope considerably. An instance of this is shown at fig. 77, but in this case the throttling is due to the throttle valve.

Of course in all engines whose speed is governed by means of a throttle valve there will always be a considerable drop of pressure between the boiler and the cylinder,

except when the engine is loaded to its full power, as the throttle valve will then be wide open. Hence if steam-chest diagrams are being taken with a view of ascertaining whether there is any undue loss of pressure between the boiler and the cylinder, except such as is brought about by the action of the governor on the throttle valve, this valve should be held or fixed wide open whilst the diagrams are being taken. Diagrams from the cylinder should be taken under the same conditions, so as to show the loss of pressure between the steam chest and the cylinder.

For the diagram taken from just on the boiler side of the starting valve or throttle valve (whichever may be the nearer to the boiler) it is not usually necessary to drive

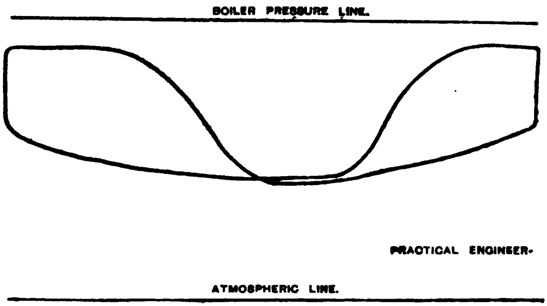


FIG. 77.—Steam chest diagram when steam is throttled.

the indicator barrel by any mechanism, but simply move it by hand, so that the pencil may mark a straight line.

After the various diagrams have been taken they should be carefully transferred to one card, so that the fall of pressure between the various parts may be more readily observed.

An instance of such a combination is shown at fig. 78. When diagrams of this character are being taken great care should always be observed to keep the boiler pressure quite steady.

Turning from the upper line to the lower line of the diagram, we find that the back pressure—that is, the pressure resisting the motion of the piston—is sometimes unduly high. In a non-condensing engine this may be brought about by the exhaust ports or pipes being of deficient sectional area

or badly arranged, by leakage past the piston or valves, or by the exhaust steam being utilised for heating or other purposes.

With a good arrangement of pipes and a well-designed cylinder there need not, in most cases, be more than 1 lb. back pressure.

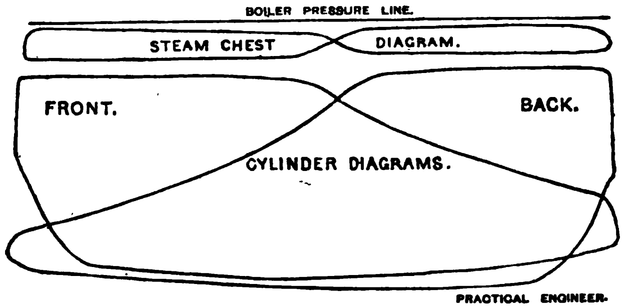


FIG. 78.—Combined diagram from steam chest and cylinder.

Fig. 79 is a diagram showing high back pressure brought about by the exhaust steam being utilised for heating purposes.

Fig. 80 shows high back pressure occurring only during

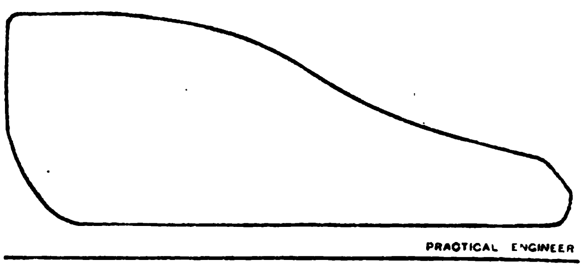


FIG. 79.—Diagram illustrating high back pressure due to exhaust being utilised for heating.

one stroke of piston. This is brought about by the exhaust port being too narrow, thus causing the back exhaust edge of the valve to come too near to the bar dividing the exhaust and front steam ports, as shown by sketch.

An examination of these diagrams shows also that the valve is set too near the front end; equalisation of the valve setting would lessen the back pressure at the front end, but would probably increase it at the back end.

One way of lessening the back pressure would be chipping as much as possible off the edges A, and then to obtain further advantage the speed of the engine might be reduced and the lap added to the steam edges of the slide valve, so as to cause an earlier cut-off.

The reduced number of revolutions would cause the exhaust to pass away slower, and thus lessen the back pressure, whilst the earlier cut-off would tend to keep the terminal pressure low, which would be advantageous as

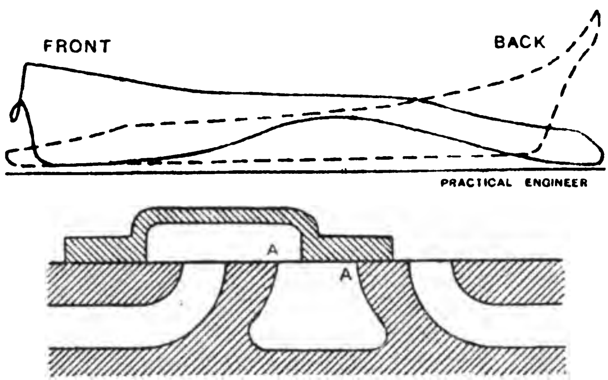


FIG. 80.—Diagrams illustrating high back pressure due to bad design of valve and unequal setting.

regards economy as well as low back pressure. Advantage might also be obtained by reducing the travel of the valve, providing the opening for steam is sufficiently large to allow of a little reduction. By this alteration the exhaust edges of the valve would be prevented from coming so close to the edges A. The small loop at the front end of the diagram is due to the lateness of admission combined with leakage of steam past the piston.

Bad vacuum in a condensing engine may be brought about in exactly the same way as high back pressure in a non-condensing engine; it may, however, be due to several other causes, such as leakage of air past the piston-rod glands, past the joints about the cylinder or the condenser, leakage

of steam past the piston, bad condition of air-pump valves, bad design of air pump, deficient supply of injection water, &c.

The last-mentioned defect is probably the most common cause of poor vacuum, and the best means of detecting it is by testing the temperature of the water in the hot well by means of a thermometer. If the temperature exceeds 120 deg. Fah., it would be well to take steps to either increase the supply of injection water or arrange for cooler injection water to be led to the condenser; either of these courses would lower the hot-well temperature, and thus be beneficial.

In all questions respecting vacuums in condensers we must never lose sight of the proportion which exists between the temperature at which water boils and the

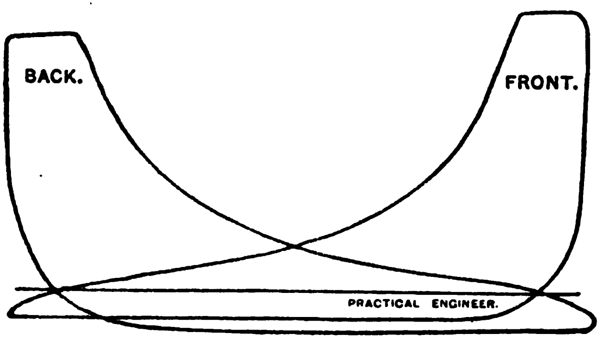


FIG. 81.—Diagrams illustrating poor vacuum at front end due to leakage past piston-rod gland.

pressure on its surface. Thus, when the pressure on the surface of the water is 3 lb. per square inch—that is, about 11¾ lb. below the atmosphere—the water would boil at a temperature of about 140 deg. Fah.; consequently, so long as this temperature was maintained the pressure could not in any way be reduced. It would, therefore, become necessary to introduce a greater quantity of injection water, so as to lower the temperature. If lowered to 120 deg. Fah., at which temperature the water would boil at nearly 2 lb. pressure, it would be impossible to obtain a vacuum of more than 12¾ lb. or 13 lb., according to the atmospheric pressure for the time being. To reduce the pressure to 1 lb.—that is, obtain a vacuum of 13¾ lb. to

14 lb.—it would be necessary to lower the hot-well temperature to about 100 deg.

When boilers are fed by water taken from the hot well, it is desirable for economical reasons that the temperature of this water be as high as possible, consistent, of course, with the conditions respecting vacuum just described; hence a compromise between the two conflicting requirements becomes necessary, and this will generally be arrived at when the hot-well temperature is about 120 deg Fah.

Where the attention to a condensing engine is not very good, leakage past the piston-rod gland not infrequently takes place, and in those cases where the piston rod passes through the front end of the cylinder this defect will often be shown on indicator diagrams by the vacuum on the front side of the piston rod not being so good as on the back. An instance of this is given at fig. 81.

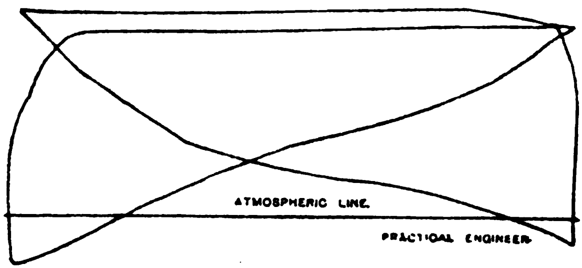


FIG. 82.—Diagrams illustrating very bad vacuum owing to throttling at exhaust valve.

It must not, however, be taken that such an inequality of vacuum always denotes leakage at the gland, as it may be also brought by local choking in the front end ports or pipes, or by leakage past the front end of the steam valve, or various other conditions; nevertheless, it is the most common cause of the inequality.

When the air leakage at a gland, or at any other part of the cylinder or condenser, is considerable, it may frequently with experience be detected by the appearance of the water in the hot well, as when air is present the water foams more than when all joints and glands are tight.

When the vacuum is poor, owing to the design of the air pump, it will often happen that the only remedy is the entire renewal of the pump, and possibly of the condenser also.

Horizontal air pumps and condensers err in this respect far more frequently than vertical air pumps. This does not appear to be due to the theory of the horizontal condenser being wrong, but rather to the fact that this style of condenser presents more difficulties of design. If arranged so that the water follows the pump piston closely, and the passages have no abrupt corners or cavities for air to accumulate in, the vacuum may equal that attained with a vertical air pump and condenser.

In all horizontal condensers the exhaust steam should be discharged into the condenser as near the top as possible, otherwise danger of water flooding into the cylinder when the engine is at rest will be much increased.

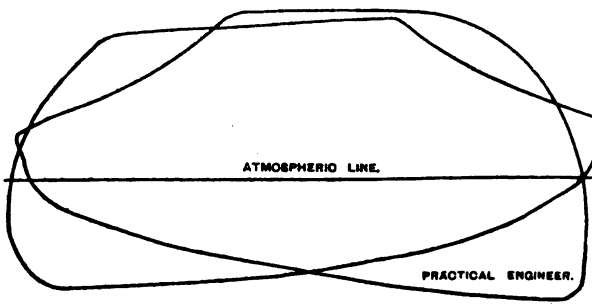


FIG. 83.—Diagrams illustrating bad vacuum owing to late exhaust opening.

An unusually bad case of deficient vacuum is shown at fig. 82. These diagrams were taken from an old beam engine fitted with drop valves of the double-beat type, and at once showed the exhaust steam was being throttled somewhere. On examination it was found that, owing to wear in the valve gear, the valves used for the exhaust were but slightly lifted, with the result that the steam had to escape to the condenser through a very small opening ; hence the excessive throttling. On increasing the lift of the valves, the vacuum was much improved, and the engine required much less steam.

An instance of bad vacuum due to late exhaust opening is shown at fig. 83. These diagrams also show considerable lateness of admission. The valves in this case are again of the Cornish double-beat or drop type.

The pistons and valves of steam engines are very liable to become leaky, and to a certain extent such leakage is shown by indicator diagrams. It is, however, quite possible for leakage to go on without it being recognisable from the diagrams. In consequence of this, it is very desirable that all pistons and valves be tested as regards their steam tightness periodically, especially when it is borne in mind that such leakage is very liable to cause considerable loss.

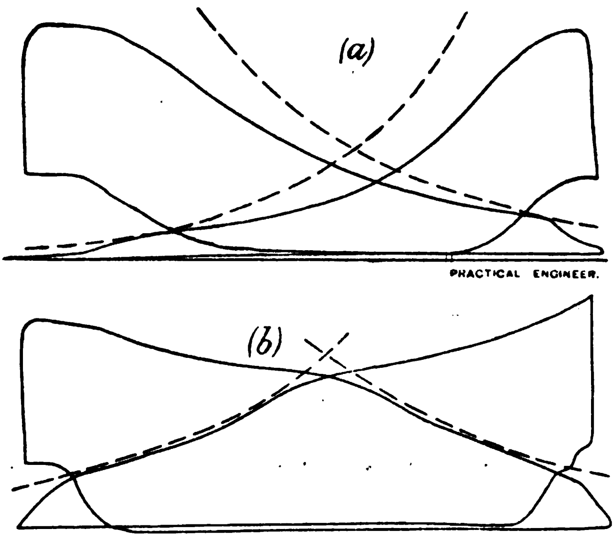


FIG. 84.—Diagrams showing leakage of steam past piston.

The piston may usually be tested by blocking the cross-head or crank securely, and then admitting the steam to one end of the cylinder. If the indicator cock at the opposite end is then opened, any steam leaking past the piston will be seen issuing from this cock, but the better course is to have the cylinder cover removed from the end opposite to that at which the pressure is applied, as the steam leaking past the piston can then be clearly seen, and there will be no mistake as to whether the steam is escaping past the piston or the valve. It is always well to test the piston at each end of its stroke, also at several intermediate positions,

as it not infrequently happens that a piston is steam-tight at one part of its stroke and not at another.

Slide valves may be tested for tightness by simply barring the engine round until the valve covers both steam ports, and then opening the starting valve, any steam leaking past the valve will be seen issuing from the steam ports, or, if the cylinder covers are not removed, it will show itself at the indicator taps.

The following diagrams indicate a few of the ways in which steam leakage is shown by indicator diagrams.

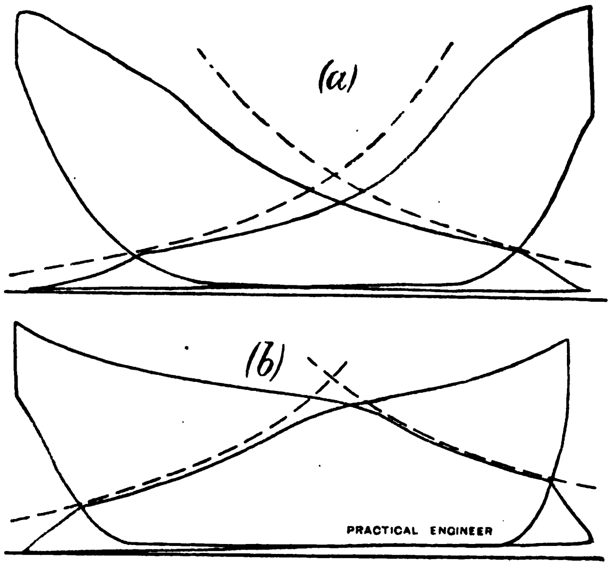


FIG. 85.—Diagrams taken from same engine as fig. 84 after stopping leakage.

In fig. 84 it will be noticed that the compression curves are not so regular as would be expected. The pistons of the engines from which these diagrams were taken were tested and found leaky when near the ends of the stroke. In one case the cylinder was re-bored, and new piston and rings put in; whilst in the other the spring rings were found to have entirely lost their elasticity, and were renewed. After these alterations, the diagrams given at fig. 85 were taken, *a* corresponding with *a* in fig. 84, and *b*

with *b*, and it will be noted that the compression curves now are as would be expected. It would appear quite possible for such compression curves as those shown by fig. 84 to be caused by excessive cylinder condensation, but the conditions under which this would occur would be very exceptional; hence, on the whole, curves of this character may be taken as showing steam leakage.

Leakage past the piston also tends to cause rounding of the admission line, similar to late admission, and may often be taken for the latter, in which case advancing the eccentric to remedy the apparently late admission would not alter the rounding of the admission corner of the diagram. Piston leakage sometimes also shows itself on the expansion curve, by causing the toe of the diagram to

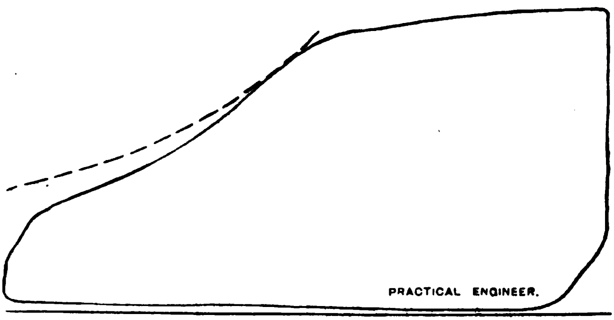


FIG. 86.—Diagram showing increased fall of pressure during expansion, owing to leakage past the piston.

fall below the hyperbolic curve, drawn to touch the expansion curve, as per fig. 86. This is due to leakage causing the pressure in the cylinder to fall more rapidly during expansion than would otherwise be the case. This might be expected to occur in all cases of piston leakage, but this is not so, as it may happen that the loss through leakage past the piston may be compensated for by leakage past the valve. In the case of compound engines, leakage past the high-pressure piston may cause the low-pressure diagrams to show a greater weight of steam than those taken from the high-pressure cylinder, but, of course, this result may also be brought about in other ways, such as leakage past a blow-through valve.

The general conclusion, then, as regards leakage past pistons is that such leakage may be shown on diagrams—(*a*)

by the shape of the compression curve; (*b*) by the rounding of the admission corner, except when this is due to lateness of admission; (*c*) by the toe of the expansion curve falling lower than would be expected on comparison with a hyperbolic curve; (*d*) by the low-pressure cylinder showing a greater weight of steam than the high-pressure cylinder. It is, however, quite possible for considerable leakage to go on without being shown by diagrams; hence, as has already been stated, pistons should be tested at regular periods.

Leakage past valves is also not always clearly shown by diagrams, as, for instance, when the steam and exhaust valves are separate and both leak, as the leakage past the steam valve would then compensate for the leakage past the exhaust valve, with the result that the pressures in the

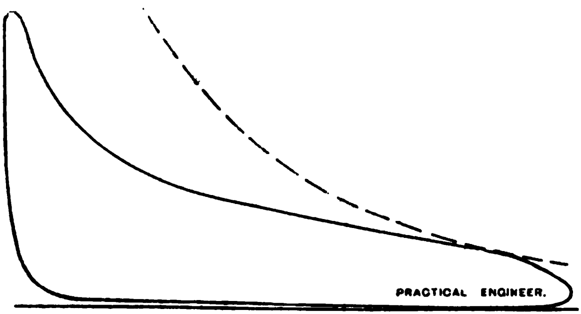


FIG. 87. Diagram showing undue pressure at toe of diagram, owing to leakage past valve.

cylinder, and shown by the diagram, would practically be the same as though no leakage occurred.

Leakage past steam valves usually causes the toe of the expansion curve to rise higher than would otherwise be the case. An instance of this is given at fig. 87. Such leakage also tends to cause increased back pressure, but as a rule the increase is small, and it would be difficult to tell from diagrams whether it was due to throttling in the exhaust pipes or to leakage past the valve.

We have now considered how most of the common defects on engines are shown on diagrams; but before commencing the next section—viz., the consideration of diagrams which are defective in themselves—we might, with advantage, look at several instances of peculiar or uncommon defects in diagrams.

At fig. 88 are shown two pairs of diagrams taken from a compound engine. It will be noted that there is a peculiar rise of pressure at the toe of each low-pressure diagram. This is due to the low-pressure valve being quite without lap, and thus not cutting off until the end of the stroke, with the result that the high-pressure exhaust opens before the low-pressure cuts off, and thus there is a fresh supply of steam sent to the low-pressure cylinder just near the end of the stroke.

With engines driving rolling mills and machinery of a similar character the load on the engine is subject to extreme changes: at one time the full load may be on, and a moment afterwards there may be no load on the engine beyond its own friction. The result of this is to cause great and rapid

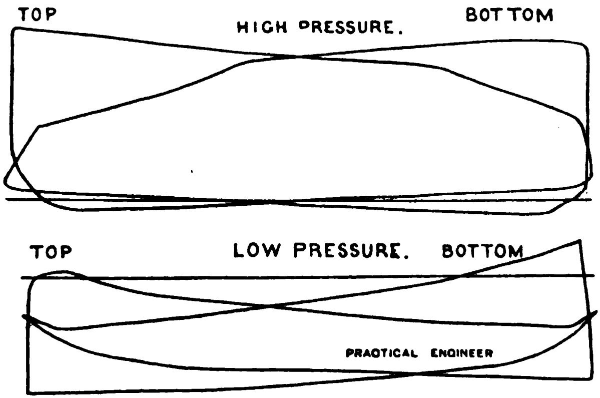


FIG. 88.—Diagrams illustrating re-admission to low-pressure cylinder before cut-off.

variation in the pressures in the cylinders, especially in the high-pressure. At fig. 89 are shown several diagrams (*a*, *b*, *c*, *d*, *e*) from a rolling-mill engine working under the conditions described above. In each the pressures tending to push the piston forward are shown in full lines, whilst the back-pressure lines are shown dotted. From *b* and *c* it will be seen that at some times the resisting pressures in the high-pressure cylinder are actually higher than the accelerating pressures. This unusual result is due to an extremely early cut-off occurring in the high-pressure cylinder so quickly after a late one that the pressures throughout the expansion in *c*, and during the greater portion of it in *b*, are

below the pressure of the steam left in the pipe between the cylinders from the last previous discharge from the high-pressure cylinder; consequently when the exhaust port is

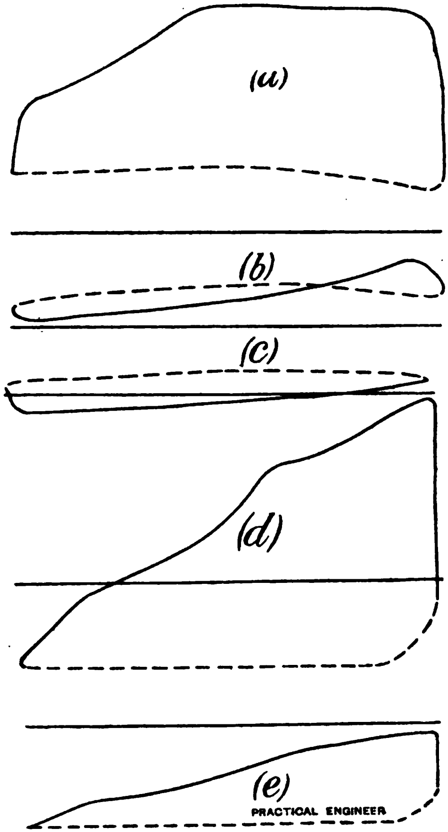


FIG. 89.—Diagrams showing effect of very irregular load.

opened there is a rush of steam into the cylinder instead of out of it.

Diagrams *d* and *e* are from the low-pressure cylinder, and illustrate the variations of pressure in that cylinder.

CHAPTER VII.

TURNING to diagrams which do not correctly show the action of the steam in the cylinder, we find that the ways in which this may occur are very numerous, and sometimes not easily detectable; the consequence is that when taking diagrams, great care should be taken in the manipulation of the indicator itself, and the diagrams should always be carefully scanned immediately after taking, and those from opposite ends of a cylinder and from different cylinders of the same engine compared, so that any inaccuracy visible may be put right or fresh diagrams at once taken.

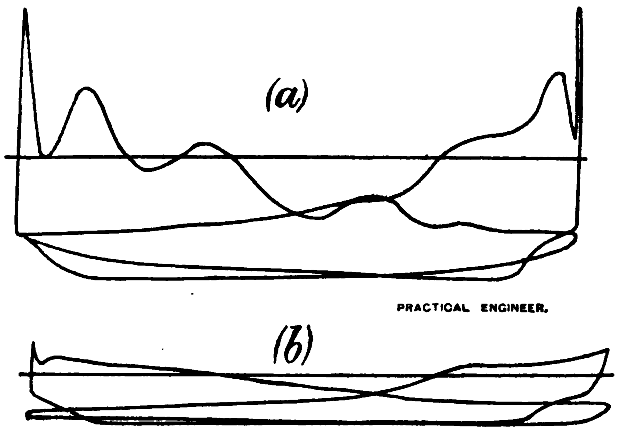


FIG. 90.—Diagrams illustrating vibration of indicator pencil owing to weak spring.

Probably the most common incorrectness in diagrams is waviness of the steam line. The cause of this was gone into fully when dealing with the different forms of indicator; but it might be well to repeat that in most cases this is due either to the indicator being of unsuitable construction for the speed at which the engine works, or to the spring used in the indicator being too weak.

At *a*, fig. 90, is shown a diagram taken with Richards indicator, at a speed of 110 revolutions per minute, and with a $\frac{1}{12}$th spring, whilst at *b* is shown another diagram taken from the same engine with the same indicator, but

with a $\frac{1}{24}$th spring, the result being that in the latter case vibration is almost absent; hence, whenever diagrams show much vibration of this character, a fresh set should be taken with a stronger spring. The use of a too weak spring sometimes also results in the initial pressure not being properly recorded, owing to its being beyond the limit of the spring. A case of this is shown at fig. 91, *a* being the diagram taken with the weak spring, *b* that with the stronger spring. Diagram *a* might easily be considered as a good one, showing little wire-drawing, and sharp cut-off; but the diagram taken with the stronger spring shows that the form of the steam lines in *a* is quite misleading.

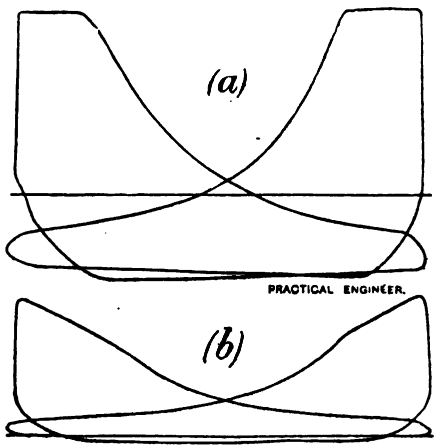


FIG. 91.—(*a*) Diagrams taken with a too weak spring. (*b*) Diagrams taken with a suitable spring.

In deciding as to the spring to use for any indication, it is necessary to not only consider the probable initial pressures, but also whether the pressures in the cylinder are likely to fall below atmosphere, as, if so, the spring used must be, if possible, such as will register to 15 lb. below atmospheric pressure, as well as the requisite amount above.

The fact that all springs are not made suitable for registering pressures below the atmosphere is often overlooked, with the result that the vacuum line is incorrect, in the manner shown by fig. 92, *a*, whilst *b* is a copy of the

diagram taken with a spring of the same scale, but of such length as to permit the indicator piston to fall sufficiently ow.

It sometimes happens, especially with high-pressure condensing engines, that there is no spring to hand which is capable of registering the required pressure above the atmosphere and at the same time registering vacuum. When this occurs, probably the best way is to take two diagrams from each end, one with a spring strong enough to correctly measure the pressures above the atmosphere, the other with a weaker spring capable of showing the

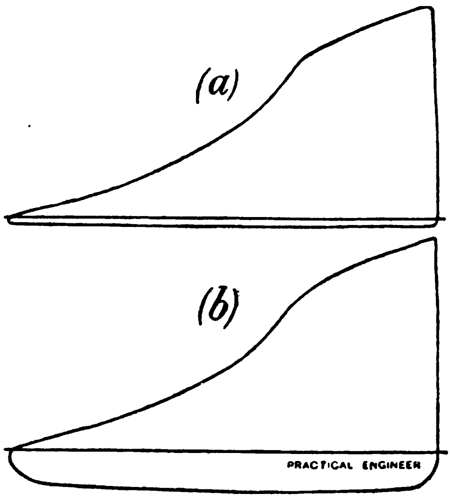


FIG. .—(*a*) Diagram taken with spring unsuitable for vacuum. (*b*) Diagram taken with spring suitable for vacuum.

vacuum correctly. In the latter case it is, of course, necessary to adopt some means of preventing the weak spring being overpressed, and this is usually done by putting a brass ferrule of suitable length on the piston rod.

An example of such combining is shown at fig. 93, *a* being the diagram taken with the strong spring, *b* that taken with the weaker spring, *c* the combination of the two to one scale.

Before leaving the question of indicator springs, it may be well to note a defect which may be caused by the manner in

which the spring is put in the Richards indicator. The spring should first be screwed on the piston and then on

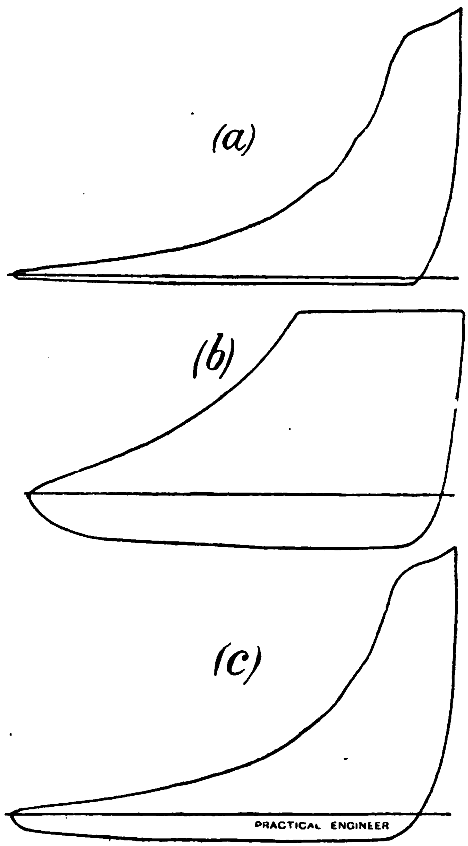


FIG. 93.—Diagrams illustrating method of using two springs, in absence of spring capable of registering the correct pressures both above and below the atmospheric line.

the cap, in each case tightening well in position, but of course not so much as to strain the spring; after this the

piston, spring, and cap should be screwed in position, and the small milled nut then attached. If the spring is loosely fixed in position instead of securely, as just advised, there is danger of the spring or piston being unscrewed, owing to the milled nut being screwed round too much, with the result that the nut may be taken so near to the cap as to prevent the piston descending sufficiently far to show the vacuum. This point will be readily understood if the reader takes a Richards indicator and simply screws the spring loosely on the piston or cap, as there will then be difficulty in saying when the milled nut is screwed home; hence the risk of

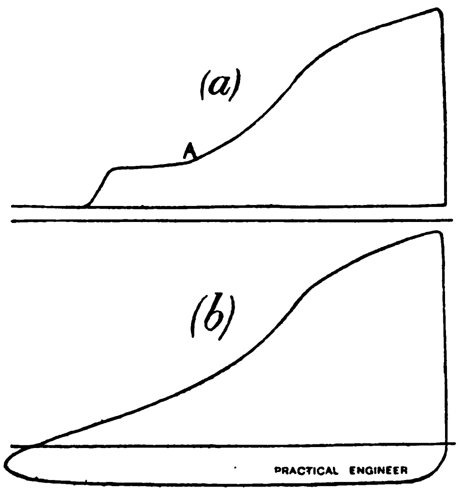


FIG. 94.—(a) Diagram caused by parallel motion catching on curved arm (b) Correct diagram.

screwing it too far down. Diagrams taken under these conditions are generally very similar to fig. 92*a*.

A somewhat similar defect is occasionally brought about in Richards indicators, owing to there being so little clearance between the parallel motion levers and the curved arm, that when the motion work joints have become worn, slight pressure of the pencil on the paper causes the parallel motion to catch on the curved arm. An instance of this is shown at fig. 94*a*. At A in this diagram, the bottom of the lever in which the pencil holder was screwed caught

on the projecting curved arm, and held there for a little while; at *b* it slipped off, and the head of the screw caught on and held there until lifted off by the steam pressure. Fig. 94*b* shows the correct diagram taken with another indicator, as it was found that with the original indicator sufficient pressure could not be put on the pencil for it to mark a distinct diagram without it catching as described.

Very untrue diagrams are frequently brought about owing to the piston of the indicator not being properly free. This is generally due to the person using the indicator

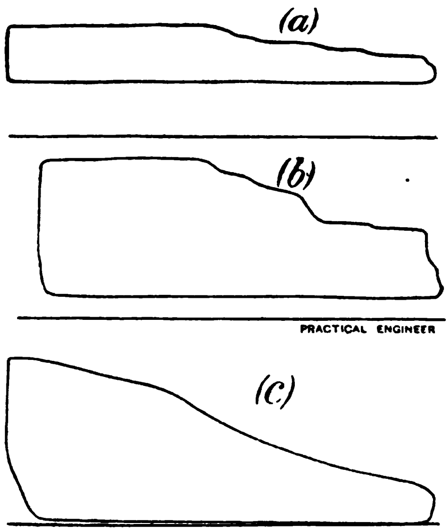


FIG. 95.—(*a*) and (*b*) Diagrams taken with indicator piston sticking. (*c*) Diagram taken with piston in free working order.

neglecting to clean it properly or sufficiently often, or in using an unsuitable lubricant. Fortunately such sticking can usually be readily detected from the diagrams; hence there is not much danger of it being overlooked.

A pair of diagrams, of which one is shown at fig. 95*a*, were taken by a firm's engine-man, and submitted for advice, but it was pointed out that the diagrams did not appear reliable; consequently fresh diagrams were taken (95*b*). These, however, were but little better, and in consequence

an experienced inspector was sent to indicate, with the result that totally different diagrams were obtained, and that shown at 95*c* was taken with the same indicator as the two immediately preceding it, but after thoroughly cleaning.

A further but rather less marked instance of sticking of an indicator piston is shown at fig. 96, *a* being the defective diagram, and *b* the correct one.

As has been already mentioned, some engineers adopt the practice of pressing rather heavily on the indicator pencil when vibration is noted, so as to reduce the vibration. This, however, should not be done, as the effect is to increase the

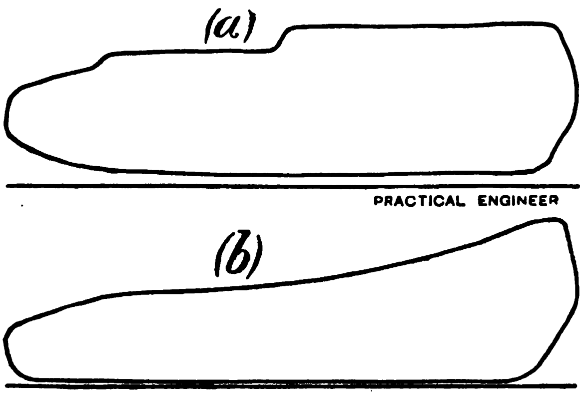


FIG. 96.—(*a*) Diagram taken with indicator sticking. (*b*) Diagram taken with indicator free.

friction of the indicator, and thus render the diagrams less reliable.

A considerable proportion of the arrangements adopted for reproducing the motion of the piston, but to a less degree, are considerably inaccurate, and the incorrectness of indicator diagrams thus caused is, as a rule, not detectable from the appearance of the diagrams; hence the greater necessity for careful observation of the reducing motion when indicating an engine. It does, however, sometimes happen that the gear is so inaccurate as to totally alter the appearance of the diagrams, as, for instance, the diagrams shown at fig. 97, which are from a horizontal single-cylinder

engine of modern construction. At the time of the indication the engine was working non-condensing, owing to scarcity of injection water.

A little study of these diagrams will show that they cannot represent the action of the steam in the cylinder. For instance, the back-pressure line of each diagram falls below the atmosphere, although the steam was being discharged direct into the atmosphere. This, of course, is impossible ; hence the natural deduction that the diagrams are incorrect in themselves, and that their peculiar form is not due to bad valve-setting or other defect in the engine itself.

The reducing motion consisted of a small crank fixed to the tail end of the crank shaft, fig. 99; in reality, this crank was simply formed by screwing a pin into the end of the shaft.

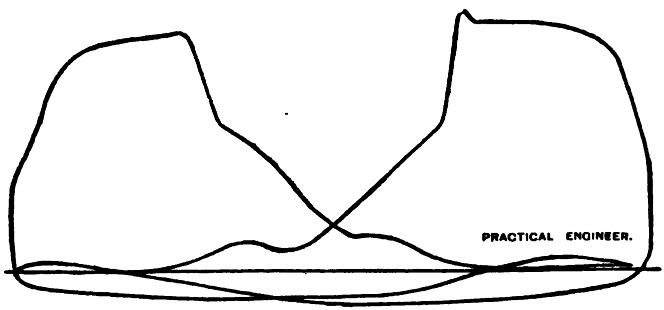


FIG. 97.—Inaccurate diagrams, owing to reducing motion being wrongly arranged.

In view of the inaccuracy of the diagrams and the type of reducing motion, the idea that the pin from which the indicator barrel was driven was incorrectly placed at once suggests itself; and to test this hypothesis, a diagram was constructed on the assumption that the points A (fig. 97) should have occurred at the ends of the stroke of the paper drum. This was done in the manner shown at fig. 98, with the result that the lower diagram was obtained, and this diagram at once explained the cause of the pressures below the atmosphere, these being simply due to the very early cut-off, consequent on the light load driven by the engine.

On investigation, it was found that the afore-mentioned hypothesis as to the cause of the peculiar diagrams was quite correct, as the cord was led away in the direction shown by sketch, fig. 99, while the pin had been fixed in line

with the crank, thus resulting in the paper drum of the indicator being nearly at the middle of its stroke when it ought to have been at the end.

Of course the pin should have been fixed so as to be in line with the cord and the centre of the crank shaft when the crank was in line with the connecting rod, as shown at *a* in

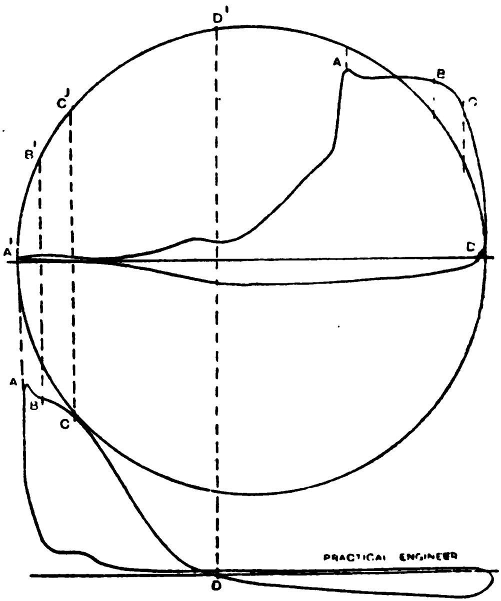


FIG. 98.—Sketch showing method of correcting approximately for the error in reducing motion, by which the diagrams in fig. 97 were obtained.

fig. 99. The motion would even then not be quite accurate, as the effect of the angularity of the connecting rod would be omitted. Hence the owners of the engine were advised to fix a reducing motion driven direct from the crosshead pin; when this was done the diagrams shown at fig. 100 were taken. These prove that the peculiarities of the former diagrams were in no way due to defective valve setting, but entirely to the reducing gear. At the time the diagrams, fig. 100, were taken the engine was working

condensing, a suitable supply of water having been obtained in the meantime.

It sometimes happens that indicator holes in cylinders are so placed as to be momentarily covered by the piston

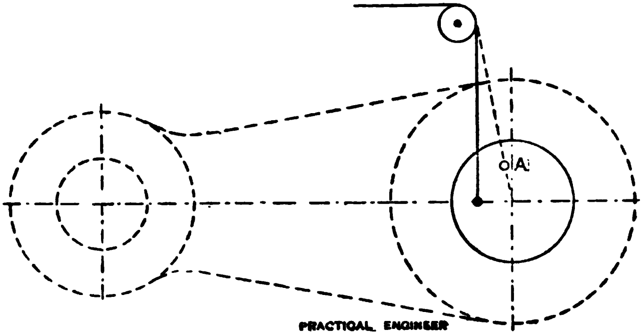


FIG. 99.—Reducing motion on engine, from which diagrams at fig. 97 were taken.

when at the end of its stroke. When this is the case the connection between the indicator and the cylinder is for the time being broken off, and consequently the diagrams do not for this period show the pressure in the cylinder. If

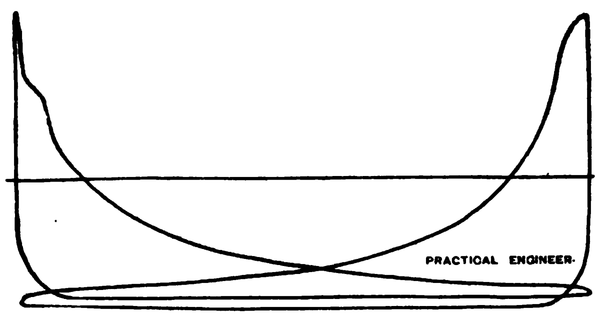


FIG. 100.—Diagrams taken from same engine as fig. 97 after altering reducing motion.

there were no leakage past either the indicator piston or the engine piston, the pressure of the steam in the indicator hole and in the indicator itself would remain constant, and a short straight line would be traced on the diagram from the

point at which the engine piston covered the opening to the end of the stroke, and this line would be re-traced until the piston uncovered the hole. This, however, does not occur in practice, as there is always fairly free leakage past the indicator piston, and often past the engine piston itself, with the result that loops of various forms are produced on the diagrams. Two instances are shown at fig. 101.

Another common cause of diagrams not representing correctly the pressures of steam in the cylinder is choking of the passage leading to the indicator, and this is especially liable to occur when the two indicator holes are connected together by means of a pipe to the middle of which the

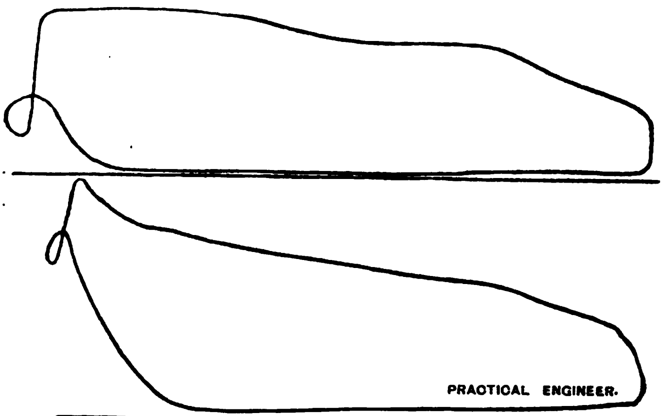


FIG. 101.—Loops at admission end of diagram, owing to engine piston coverin indicator hole.

indicator is attached, as experience shows that grease and other matter are liable to accumulate in such pipes, unless regularly used. Hence it is advantageous, although less convenient, to attach a separate indicator tap to each end of the cylinder, and connect the indicator directly to each end. Sometimes the substitution of a loop or connecting pipe such as just described by separate direct taps causes a very material improvement in the diagrams, especially as regards vacuum, and the writer has known cases where owners of engines have been at considerable expense in trying to find the cause for the great difference between the vacuum in the cylinder, as shown by the diagrams, and the vacuum in

the condenser as shown by the gauge, only to find ultimately that the difference has been due to inaccurate diagrams from the cause just explained.

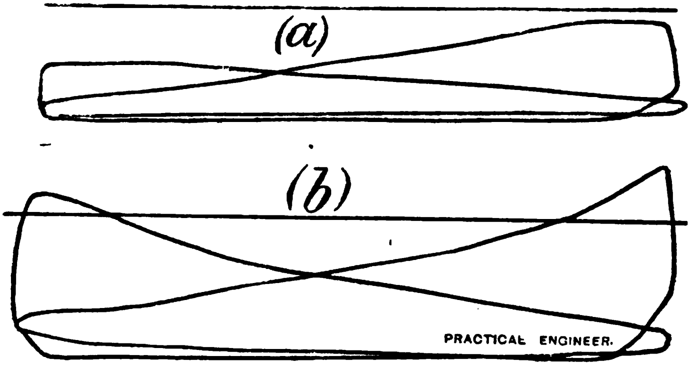


FIG. 102.—Diagrams showing effect of choking in pipe connecting the opposite ends of the cylinder; diagrams (a) being taken with pipe partially choked, and (b) after substitution of pipe by separate taps at each end of cylinder.

It must not be understood from the remarks just made that loop pipes are in all cases objectionable. This is not the case, as various careful experiments have shown that, if

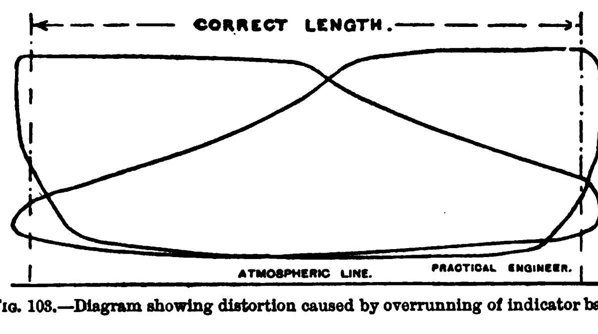


FIG. 103.—Diagram showing distortion caused by overrunning of indicator barrel.

kept thoroughly clean internally, the diagrams for strokes of ordinary length are satisfactory. The objection to such pipes is simply the danger of grease or other matter accumulating in them, and causing choking of the steam

passing to and from the indicator, and such accumulation is especially liable to occur with engines which are very seldom indicated.

When dealing with the various types of indicator in the first chapter, reference was made to the danger of inaccurate diagrams being produced at high speeds owing to the

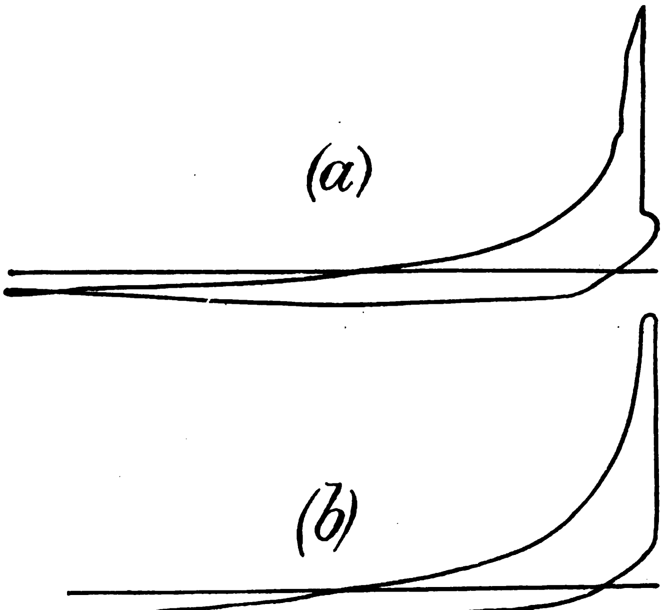


FIG. 104.—(a) Diagram showing effect of using cord of an elastic character.
(b) Diagram taken with cord of a suitable character.

inertia of the indicator barrel. It will therefore not be necessary to deal with this matter at all fully here, but to facilitate reference the diagrams shown at fig. 12 are reproduced at fig. 103.

Attention has also been called previously to the importance of using cord of an inelastic character for driving the indicator barrel, as, owing to the pull varying considerably, the cord will certainly vary materially in length if it is of

material that stretches readily; hence special cord should always be used, and it is desirable even then that the cord be stretched by a weight or other arrangement for a few days before it is used. An illustration of the effect of an elastic cord is shown at fig. 104*a*. In this case the cord used was the ordinary cotton string used largely in cotton mills, and was of considerable length. The correct diagram is shown at fig. 104*b*.

The diagrams shown at fig. 105*a* are reproductions to a reduced scale of diagrams taken from a rather small vertical

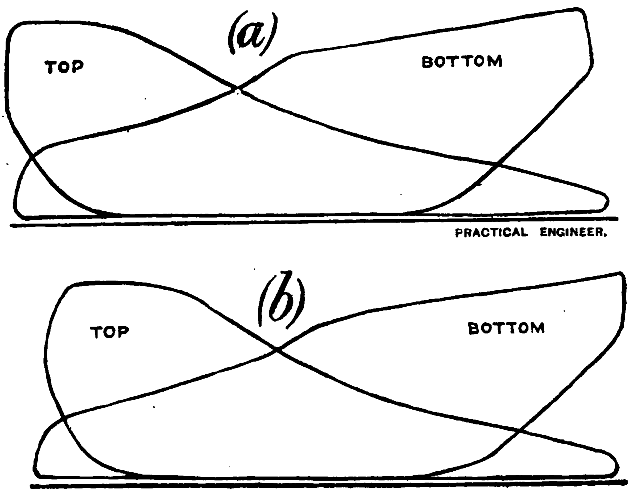


Fig. 105.—Diagrams illustrating inaccuracy due to paper drum not being parallel with indicator cylinder.

engine, fitted with a plain slide valve. From them it is evident that the valve was unequally set, and was too near the top end, thus causing the points of cut-off, exhaust opening, and exhaust closing to be unequal on the opposite sides of the piston. In view of the fact of the valve apparently being so much nearer the top than the bottom, it would be expected that the admission to the bottom end would be materially earlier than to the top, but from the diagrams this would not appear to be the case; hence the natural conclusion that the diagrams did not truly represent

the action of the steam in the cylinder. On investigation it was found that the spindle of the paper drum of the indicator had been bent, with the result that the drum was not parallel with the indicator cylinder, and, consequently, lines traced by the vertical movement of the pencil were not perpendicular with the atmospheric line, thus giving an appearance of late admission at one end, whilst at the opposite end the admission line was more nearly vertical than it should have been.

By reconstructing the diagrams so as to allow for the inclination of the paper drum, diagrams, 105*b*, were obtained. From these it will be seen that the inequality of valve setting was hardly so great as would appear from fig. 105*a*, and that the admission to the bottom was not late.

The defect just described shows the necessity for careful examination of indicators at regular periods, as the bending of the drum spindle had been, in all probability, caused by a fall of the indicator at some time.

The following is a simple method of testing the paper drum: Remove the spring from the indicator, and move the pencil round until its point just touches an indicator paper wrapped tightly round the drum; then raise the pencil upwards by means of the nut on the top of the piston rod, and note if the pencil touches the paper evenly throughout its movement, and do this for several different positions of the paper drum. Then, with the pencil in its lowest position, mark a horizontal line on the paper, and afterwards, with the drum fixed in several different positions, mark vertical lines on the paper, after this remove the paper and test, by means of a square or other convenient method, whether these lines are strictly perpendicular to the horizontal line. If not, or if the pencil does not bear evenly all over the paper, the drum should be adjusted.

As has already been explained, there is considerable difficulty in making reliable indicator springs, and this is not the only difficulty in connection with such springs, as, even when made accurate in the first instance, they are liable to change, and thus become more or less inaccurate by use. This being the case, it is essential that, in all cases where it is necessary the pressures in the cylinders be determined closely, the springs used in the indicator be tested from time to time. Most frequently such springs are tested cold by means of weights. There is, however, an objection to this course, as experience has shown that the strength of springs varies a little according to their temperature; hence a better method

is to test the springs by means of steam pressure, and compare the pressures registered by them with the pressure registered by, say, a pair of carefully-tested gauges, the gauges being, of course, attached to the testing boiler by suitable syphons, in such a manner as not to be heated by the steam, and thus remain at the temperature at which they were tested. This method of testing indicator springs has a further advantage over the system of testing by dead weights, as it tests the whole combination—that is, the area of the piston, the parallel motion, and the spring together, instead of the strength of the spring alone.

Where the spring is tested separately, the parallel motion and the diameter of the piston should also be tested by separate methods.

Although the method of testing springs by steam pressure seems preferable to the cold testing by weights, it is, however, probable that even it is not quite accurate, as there is doubt as to the actual temperature of the spring during use in the indicator, owing to the fluctuations of steam pressure, and therefore of temperature, being so rapid that it hardly seems probable the temperature of the spring will keep pace with them.

We have now considered a large number of diagrams, which either show that the engine to which they relate is not in proper order, or which are incorrect in themselves, and from the methods adopted in connection with these it is possible that readers will have little difficulty in determining the causes of most of the peculiarities shown by diagrams brought under their notice.

It might not be out of place here to say that in no case should a reader endeavour to commit to memory the various forms of the diagrams showing different defects, but he should practice thoughtful analysis of diagrams, and use the illustrations given as examples of the mode of reasoning which may be adopted.

Before leaving the subject, a few words in reference to triple and other multiple expansion engines seem desirable, but as these would be out of place in this chapter devoted to defective diagrams, the matter will be carried into a separate chapter.

CHAPTER VIII.

Multiple-expansion Engines.

Triple-expansion engines—that is, engines in which the steam is expanded in three cylinders—have been found to give more economical results than either single or two cylinder engines, whilst in some cases quadruple-expansion—that is, four-cylinder—engines have been shown to be still more economical.

It is not intended in this chapter to enter into a discussion as to why these multiple-expansion engines are more economical than single-cylinder engines; but it may be mentioned that the improvement is generally considered, and probably truthfully, to be mainly due to the fact that with several cylinders the range of temperature in each cylinder is less than if the steam were expanded to the same extent in only one cylinder, thus causing diminished loss by cylinder condensation. The manner in which this is brought about was explained in connection with compound engines, and a diagram (fig. 53)* was used to illustrate it; also the great losses caused by cylinder condensation where the degree of expansion is high, were discussed in the same chapter, and illustrated at figs. 51 and 52.

Although expansion of steam in several cylinders is advantageous as regards cylinder condensation, it must be remembered that the additional cylinders introduce some disadvantages, as the largely increased surface doubtless materially increases the loss by radiation of heat; also there are losses owing to drop of pressure between cylinders; friction of the steam passing through the additional ports and passages; mechanical friction of the extra pistons, valves, and other working parts. Taking the foregoing into account, it will readily be seen that the gain by lessened condensation must be considerable before it counterbalances the losses introduced by the additional cylinder or cylinders, and that if the degree of expansion is small, it is quite possible for a single-cylinder engine to be more economical than a compound, and a compound more economical than a triple-expansion engine.

Speaking generally, it has been found that for steam pressures below 50 lb. to 60 lb. per square inch single-cylinder engines are preferable; for pressures between 60 lb. and 120 lb. per square inch two-cylinder compound

engines are desirable; while triple-expansion engines are generally adopted when the pressure is between 120 lb. and

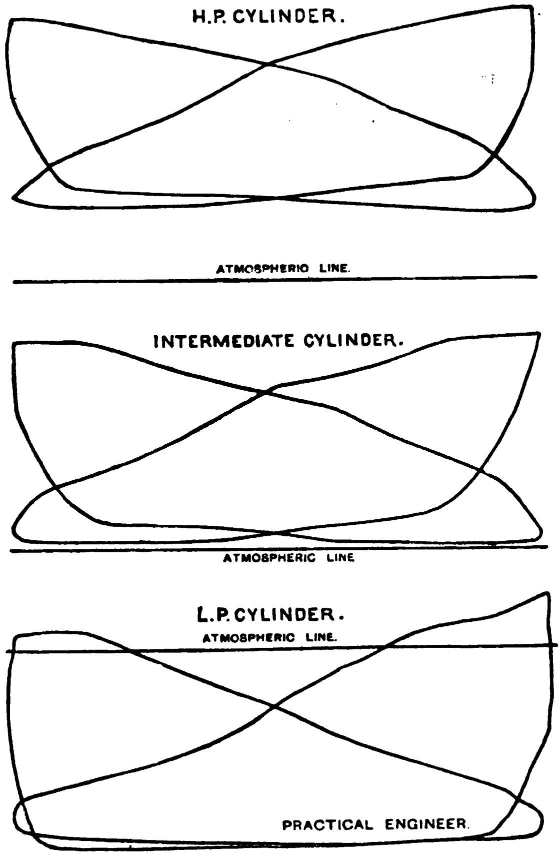


FIG. 106.—Diagrams from ordinary triple-expansion marine engine.

200 lb. per square inch. Quadruple-expansion engines are now rarely put down for pressures less than 180 lb. per square inch.

Apart from improvement due to lessened cylinder condensation, multiple-expansion engines tend to be more economical than single-cylinder engines, owing simply to the increased steam pressures, as with any engine there must necessarily be some loss by incomplete expansion and back

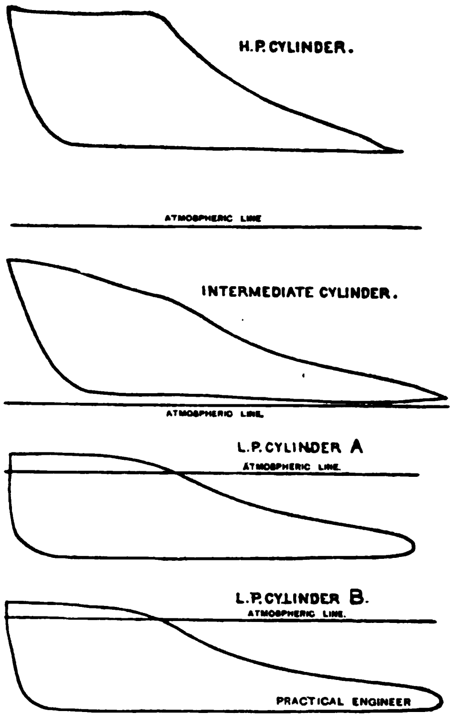


FIG. 107.—Diagrams from triple-expansion engine.

pressure ; and if the pressure is low, this loss will represent a large percentage of the total work done by the steam, whilst this percentage will get lower as the initial steam pressure increases.

As regards actual diagrams, those taken from both triple and quadruple expansion engines will, of course, follow the

general principles of those from simple engines, and, as regards defects, will bear the same form of reasoning.

An instance of the type of diagrams commonly obtained

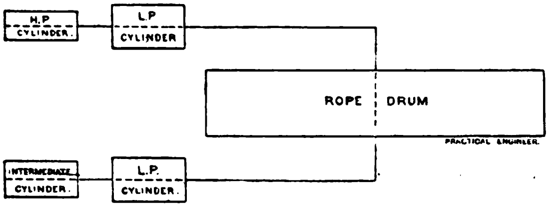


FIG. 108.—Rough plan showing arrangement of cylinders of engine from which diagrams at Fig. 107 were taken.

from triple-expansion engines is given at fig. 106. Another set of diagrams from a triple-expansion engine are given at fig. 107. In this latter case there are two low-pressure diagrams

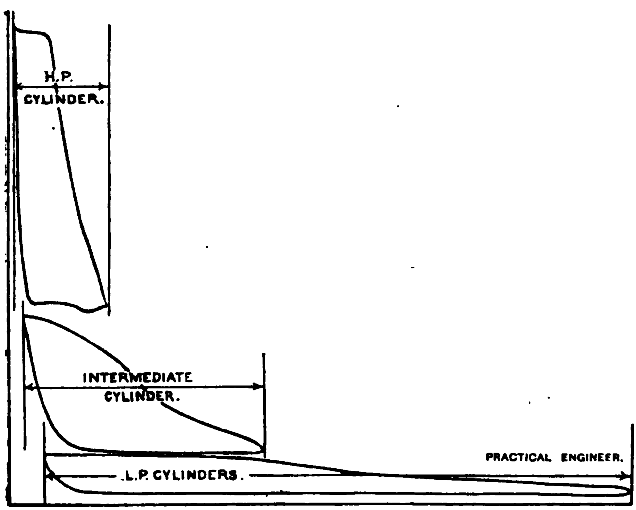


FIG. 109.—Combination of diagrams shown at Fig. 107.

owing to there being two low-pressure cylinders, in accordance with the plan now usually adopted for large triple-expansion mill engines. The arrangement of the cylinders

is indicated by the rough plan at fig. 108, the high-pressure and one low-pressure cylinder being placed tandem on the left side, whilst the intermediate and the other low pressure are similarly placed on the right side, the cranks being fixed at right angles. These diagrams are combined at fig. 109, the method of combining being exactly the same as was adopted for compound engines, which latter have already been fully described.

Owing to there being two low-pressure diagrams, it is, of course, necessary, before commencing combining, to construct a mean diagram for the low-pressure cylinders. A simple method of doing this would be to trace each diagram on one atmospheric line, and then draw a line midway between the two diagrams wherever they do not coincide.

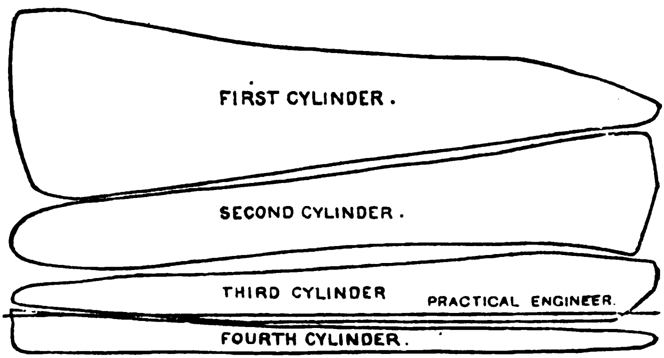


FIG. 110.—Diagrams from Adamson's quadruple-expansion engine.

At fig. 110 are shown a set of diagrams taken from a quadruple-expansion engine made by Messrs. Daniel Adamson and Co., in 1872. The writer believes that this engine was the first quadruple-expansion engine made. Mr. Adamson took out a patent for triple and quadruple expansion engines early in 1861, and constructed a triple-expansion engine in the same year, but a considerable time elapsed before the system became at all general. The quadruple-expansion engine in question was worked from boilers whose safety valves were loaded to 110 lb. per square inch. So far as the writer is aware, no reliable tests were made as to the steam consumption per I.H.P. of this engine, but the coal was tested several times. The average

weight of coal used per hour, as given by Mr. Adamson in a paper he read before the Iron and Steel Institute, was 1·77 lb. per hour. This consumption is no better than has been since obtained with compound engines, very probably owing to the boiler pressure being too low for expansion in four different cylinders, and not in any way due to any fault in the system or the engine itself, for, as has been already stated, quadruple-expansion engines have been found, with boiler pressures of over 180 lb. per square inch, to be more economical than any other type at present in use.

TESTING OF ENGINES AND BOILERS.

CHAPTER I.

WITHIN the last few years the practice of making complete economic tests of engines and boilers has extended very greatly. The result of this will doubtless be that reliable and accurate information of most of the various types of engines and boilers will before long be accessible for engineers, millowners, and other persons, to whom such information cannot help but be of great value. In view of the extension just mentioned, it is probable that many engineers who have not had previous experience in this direction will find it necessary to undertake the testing of some portion or portions of a steam plant. To these the writer hopes that the following notes will be of service.

The owner of a mill or works would seldom think of going to the expense or trouble of a complete test of his engine and boilers, and, except under unusual circumstances, it would be sufficient for him to know what weight of coal he is burning per horse power developed. In a very large number of cases even this is rarely ascertained.

With this method of comparing only the weight of fuel used with the power developed the owner is usually satisfied, providing the resulting figure for the number of pounds of coal per I.H.P. is about the same as before, and rarely is a systematic attempt made to ascertain whether the figure might be lowered. Again, the figure itself will be very often materially inaccurate, as it is quite customary to base the calculation on one set of diagrams, which are very often taken when the heaviest load of the week is on the engine, instead of when a fair average load is being driven. This system unfortunately causes very misleading numbers to be spread abroad in reference to the working of engines and boilers, and it possesses the great disadvantage of not separating the working of the boiler from that of the engine; consequently, it not infrequently happens that the engine may be kept in excellent condition, and be working very favourably, whilst the boiler is being fired badly, or

not properly attended to, or *vice versa*. The result may be that the whole combination may be working moderately, instead of very economically.

Instead of the system just mentioned, it would be much better, especially where the engines are large or coal very dear, to adopt some means of measuring the weight of water used per I.H.P., as well as the coal.

A method of measuring water by means of tanks will be fully described further on. This arrangement, although the most accurate, is somewhat troublesome, and requires constant attendance; it is, however, the only one which should be adopted where strictly accurate results are required. For approximate results, obtainable by a millowner with his ordinary staff, a good water meter may be used. Such a meter, fixed on the feed pipe leading to the boilers or to the economiser (if there is one), would enable an owner to ascertain approximately the amount of water used in any given time.

Where there is an economiser or other form of feed-water heater, the meter should be placed so as to measure the water before it enters the economiser or heater, as at high temperatures the risk of inaccuracy is very much greater.

By means of a meter, arranged as described, it would be possible to compare easily the comparative value of the different classes of coal used, as the amount of water evaporated per ton of each kind of coal could easily be noted; also, if the boilers were fired at different times by different firemen, it could easily be seen whether the system of firing adopted by one man gave better or worse results than that given by another. One firm who adopted a system similar to that just described found that a difference of 10 per cent had frequently occurred with different firemen.

Again, by measuring the water as well as the coal, the figures obtained would give the number of pounds of water evaporated per pound of coal, and the number of pounds of steam used per I.H.P. by the engine, so that the efficiency of the engine and the boiler would be obtained separately instead of together. The value of this will be seen from the following illustration: Suppose that a compound engine and boiler are using $2\frac{1}{3}$ lb. of coal per I.H.P. This would be a fair result, but not very good, and would probably be left alone by most millowners. If, however, for example, the $2\frac{1}{3}$ lb. of coal per I.H.P. were made up thus—10 lb. of water evaporated per pound of coal, and $23\frac{1}{3}$ lb. of steam used per

I.H.P., it would be at once seen that the engine was much at fault, and should have attention. On the other hand, suppose that the engine used 14 lb. of steam per I.H.P.; this would mean that when using $2\frac{1}{3}$ lb. of coal per hour only 6 lb. of water were evaporated per pound of coal, in which case the boiler would be much at fault.

In those cases where the engine is at fault the steps taken to improve it will, of course, vary greatly, according to circumstances, and no general mode of procedure can be laid down. The same remark would apply to boilers; but in the case of boilers it would be wise, in the first instance, and before making expensive alterations, to have samples of the gases leaving the boiler collected and analysed, so as to ascertain if the coal is being properly burnt. In many cases where this has been done it has been found that considerable loss was being caused by the amount of air passing through the furnace being greatly in excess of the amount required.

Methods of collecting and analysing flue gases will be explained further on.

So far we have only briefly considered tests of a commercial character. When we turn to the engineer, we find that for him it is very desirable, and even essential, that tests of engines and boilers, &c., be as complete in every detail as is possible, as he requires to compare the results not only with previous results of tests of the same engine and boilers, but with tests of others working sometimes under very different conditions. He also requires to know, as far as possible, the disposal of every unit of heat, so that he may be assisted in devising means to improve the plant. It is to the latter kind of test that this chapter is devoted.

For the fixing of our ideas we will assume that a plant, consisting of a triple-expansion engine, economiser, and boiler, are to be tested, and proceed at once to consider the various observations to be made, the apparatus required, the method of tabulating, and the various calculations.

CHAPTER II.

Boiler.

The following are the principal observations to be made in connection with this. This list might, of course, be extended considerably in some cases, to meet special requirements or experiments.

(a) Measurement of coal and ashes, and collection of coal samples.
(b) Measurement of water.
(c) Pressure in boiler.
(d) Height of water in boiler.
(e) Temperatures of feed water, air entering furnaces, and gases leaving boiler; also temperature in furnaces, if possible.
(f) Analysis of coal.
(g) Collection of samples of flue gases, and analysis of same.
(h) Testing dryness of steam.
(i) Height of barometer and state of weather.

(a) Measurement of Coal.

The method of weighing the coal will, of course, vary according to circumstances, but the most common is that of weighing small quantities in a barrow or box, as they are required. Where the quantity of coal to be used during a test is not large, and can be estimated fairly closely, the total amount required may be weighed out prior to the commencement of the test, and the surplus weighed back, and deducted.

If the latter plan is adopted, the time at which each furnace is fired, and the number of shovels thrown on, should be booked, so that the rate of firing at different periods of the test may be determined.

Where this plan of weighing out the coal as it is required is adopted—and it will be the best in the majority of cases—it may be carried out as follows:—

Have a weighing machine in the stokehole, and arrange a plank run to it so that a barrow can easily be wheeled on to the platform of the weighing machine. After carefully weighing the barrow, set the machine for an even weight of coal—say 2 cwt.—and weigh out sufficient coal on to the

stokehole floor to keep the firemen going for, say, half an hour, repeating the weighings as required throughout the test.

It is wise to keep, as far as possible, each fresh lot of coal separate from the lot partially used, and book the time at which each lot is commenced of and finished.

Each time ashes are raked from under the fires they should be carefully weighed, and the time booked; also whenever the fires are cleaned the ashes and clinker should be withdrawn into an iron barrow, and weighed.

In addition to weighing the coal, it is essential that the fires be so worked as to be as nearly as possible the same thickness and in the same condition at the end of the test as they were at the beginning. As regards the thickness, the only method of comparing is by the eye; as regards condition, probably the most satisfactory method to adopt is that of cleaning out the fires, say half an hour before the commencement of the test, and doing likewise half an hour before the finish.

The various observations made by each person engaged on the test should be booked down on suitable log sheets; for the coal the sheet might take a form somewhat as follows:—

COAL.

Test commenced ....
State of fires at commencement of test.................
Time fires cleaned before test ....
Times fires cleaned during test ....
Net weight of barrow...
Amount weighed each time in barrow ....
Amount taken out for sample ....

No. of barrows weighed.	Time coal commenced of.	Time finished.

When weighing the coal a handful should be taken from each lot of coal put on the floor, to form a sample, and the total amount thus collected should be deducted from the total weighed out, to give the actual weight used.

The actual quantity required for an analysis is extremely small, but, to get it thoroughly representative, a system such

as just described should be adopted. The amount thus collected should be broken in a mortar to small pieces, which should then be well mixed; a smaller sample should then be taken from this and broken finer. This course should be repeated a time or two, until a small quantity is obtained in a fine powder, and thoroughly representative of the general mass of the coal, instead of being crushed out of a single piece of coal.

Between the collection and analysis of the coal it is essential that the sample be kept in such a place that there is no danger of the amount of moisture it contains being altered, as the percentage of moisture forms an important element in the calculations to be made afterwards.

The method of analysing and of testing the calorimetric value of the coal will be referred to further on.

For the ash and clinker the following log sheet would be suitable:—

ASH AND CLINKER.

Ash from below grate, or clinker from fire.	Time withdrawn.	Weight.	Approximate temperature.

It is hardly necessary to mention that the weighing machine used should be very carefully tested before the commencement of the test, as should all other apparatus used.

(*b*) WATER MEASUREMENTS.

For the measurement of the amount of the water passed into the boilers the most satisfactory method is that of a pair of tanks of known capacity emptying into an auxiliary tank. A sketch showing the usual method of arranging such tanks is given at fig. 1.

The two upper tanks A and B are the tanks in which the water is measured, the exact capacity of the tanks up to the level of the overflow openings having been previously determined with water as nearly as possible at the same temperature as the water used during the test.

In use, tank B is filling by means of the overhead pipe, whilst tank A is emptying into the auxiliary tank or

reservoir C below, from which the boiler pump draws its water, and *vice versa.*

At the commencement of the test the level of the water in the auxiliary tank should be measured, and at the finish of

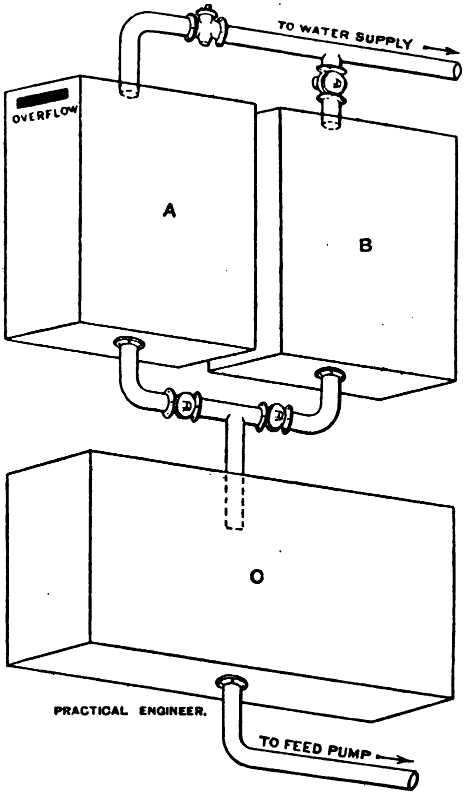


FIG. 1.—Arrangement of Tanks for Measurement of Feed Water.

the test the water should be taken to exactly the same level, which, preferably, should be high in the tank.

The temperature of each tank full of water should be ascertained and booked.

The log sheet for the water measurements might take the following form :—

WATER MEASUREMENTS.

Capacity of tank A .... gallons.
Capacity of tank B .... gallons.
Temperature at which capacities tested ....
Level of water in auxiliary tank at commencement of test .... } ....
Level at finish of test ....
Amount of water left in tank A and B at finish......

Tank A.			Tank B.		
Time commenced emptying.	Tempera-ture.	Level in auxiliary tank.	Time commenced emptying.	Tempera-ture.	Level in auxiliary tank.

It is sometimes wise to also note the speed of the donkey pump, say at half-hourly intervals, and, in addition, the steam pressure, the nature of the smoke from the chimney, the temperature of the air in firehole, and the level of the water in the boilers. For the three last mentioned the log sheet might take the following form :—

WATER LEVEL, PRESSURE, AND FIREHOLE TEMPERATURE.

Height of water in gauge glasses at start...............
Distance from bottom of gauge glasses to level of furnace crown .... } ....
(The last dimension is to enable the steam surface to be calculated.)

Time.	Height of water in gauge.	Pressure.	Temperature in firehole.

(*e*) Temperatures.

The various moderately low temperatures may, of course, be read by ordinary thermometers; but the temperatures in the flues will sometimes be higher than the temperature at which mercury boils; hence ordinary thermometers are not applicable for flue-testing. There are many kinds of pyrometer in the market, based on different principles, and intended for the registration of high temperatures, but as far as the writer is aware none of these are really reliable.

The gases leaving boilers rarely exceed 900 deg. Fah. in temperature. For temperatures not exceeding from 900 deg. to 1,000 deg. Fah., mercury thermometers with compressed nitrogen over the mercury may be obtained. The object of the compressed nitrogen is to raise the boiling point of the mercury.

Even this class of thermometer cannot, however, be considered strictly accurate, as at high temperatures the glass of the thermometer is rather liable to change, and thus alter the capacity of the bulb in which the greater portion of the mercury rests. On the whole, however, it has been generally found more trustworthy than any of the ordinary pyrometers, for temperatures not exceeding 1,000 deg. Fah.

A very convenient form of nitrogen flue thermometer is made by Messrs. Schäffer and Budenberg, of Manchester.

As regards the temperatures in the furnaces, it is to be regretted that there does not exist some convenient and reliable apparatus for the determination of these, as such information, if reliable, would doubtless be of great value.

(*f*) Analysis of Coal.

The method in which a sample of the coal should be collected has already been explained. The analysis of this sample would necessarily be done after the test by a chemist, unless the engineer making the test were capable of analysing it.

Although in the majority of cases a chemist would be employed for this work, it will not be out of place to describe, briefly, the manner in which such analyses are generally made.

The first process is to dry the coal and determine the amount of free moisture. For this purpose a known weight of coal is taken and placed in an ordinary laboratory oven heated to about 212 deg. Fah. At this temperature the whole of the free moisture is evaporated off in due course; this point is determined by weighing the coal until no

further reduction in weight occurs. The difference between the original weight and the weight after drying gives the amount of moisture in the coal. It should be noted that the temperature of 212 deg. Fah. is not much exceeded, otherwise there is danger of some of the volatile hydrocarbons in the coal being evaporated.

Having determined the moisture, a known weight of the dried coal is taken and treated for the determination of carbon and hydrogen. This is usually done in a combustion furnace, such as is used for organic analysis, the method being that of heating the coal in a closed tube to which pure dry oxygen can be supplied. The oxygen passing over the heated coal brings about combustion, the carbon of the coal combining with some of the oxygen, forming carbonic acid gas ($C\,O_2$), whilst the hydrogen combines with oxygen, and forms steam ($H_2\,O$).

These products of combustion are passed through calcium chloride drying tubes to absorb the steam, then through several bulbs containing caustic potash, to absorb the carbonic acid. By noting the increase of weight of the drying tube and the potash bulbs respectively, the amount of steam and carbonic acid resulting from the combustion of the coal is obtained, and it is easy then to calculate the percentage of carbon and hydrogen in the coal.

Means are, of course, taken to prevent any sulphur or nitrogen compounds passing to the bulbs or drying tube and causing misleading results.

The remaining substance, after the combustion just described, is ash, but, as a rule, the ash percentage is obtained separately from a larger quantity of coal, by heating in a crucible over a Bunsen burner until no further reduction of weight occurs.

The sulphur may be determined by several processes, but as these are described fully in many text books on chemistry, it is hardly necessary to enter into them here.

The nitrogen is a somewhat complex and difficult element to determine; hence, in most commercial analyses the analyst estimates this element, being guided in his estimate by known complete analyses of coals of a similar character.

No satisfactory method appears to have yet been brought forward for the direct determination of the amount of oxygen in coal, hence this has to be obtained by difference; that is to say, if the previous elements accounted for 95 per cent of the coal, the remaining 5 per cent would be called oxygen. Thus the oxygen would suffer for any errors in the previous determinations.

The composition of several different coals is given below. These are taken from D. K. Clark's "Steam Engine."

	Welsh.	Newcastle.	Derbyshire and Yorkshire.	Lancashire.	Scotch.	Foreign Coals. Van Diemen's Land.	Chili.	Lignite, Trinidad.
Carbon	83·78	82·12	79·68	77·9	78·53	65·8	63·56	65·2
Hydrogen..	4·79	5·31	4·94	5·32	5·61	3·5	5·43	4·25
Sulphur	1·43	1·24	1·01	1·44	1·11	1·1	2·5	0·69
Nitrogen ..	0·98	1·35	1·41	1·30	1·00	1·3	0·82	1·33
Oxygen	4·15	5·69	10·28	9·53	9·69	5·58	14·84	21·69
Ash	4·91	3·77	2·65	4·88	4·03	22·71	13·31	6·84

A knowledge of the composition of the coal enables us to compare the relative values of the coals used on different tests, and the tests to be made comparable on a definite basis. The standard of comparison for different coals now most generally adopted is the "carbon value." This value is the relative heating value of 1 lb. of the coal compared with the heating value of 1 lb. of pure carbon. The analysis also provides data for a number of important calculations which will be dealt with in due course.

(*g*) Collection of Samples of Flue Gases and Analysis of Same.

The manner in which samples of flue gases are collected will greatly depend on the purpose for which the analyses are intended. If simply intended to show whether or not the fuel is being properly burned, the sample may be collected by means of a bellows or other handy aspirating arrangement. If, on the other hand, the samples are to show the average composition of the flue gases, from which composition important calculations or deductions have to be made, it is desirable that the collection of such sample extend over a fair length of time, so that the sample may be more representative of the average composition.

We will first suppose that the object of the analysis of the gases is to note whether satisfactory combustion is

going on. Have a hole made in some convenient portion of the main flue; then put a pipe through the opening, the pipe being of such length and so arranged that one end is approximately at the centre of the flue. For boiler flues, the pipe may be ordinary steam or gas pipe, ¼ in. or ½ in. bore. After insertion of the pipe, the hole in the flue wall should be luted with clay to prevent air leaking into the flue. The outer end of the pipe may be at any accessible position. A small piece of indiarubber tube should then be fixed to the pipe, a little glass wool having previously been placed in the pipe to prevent any dirt being drawn from the flue. The bellows for drawing out gas from the flue should then be attached to the indiarubber tube, after which gas should be drawn into the bellows and then discharged into the atmosphere several times, so as to thoroughly remove all air from the bellows and tube. After a fair sample has been drawn in the bellows, these should be disconnected from the iron pipe, so that the sample may be transferred to the analysis apparatus. This apparatus will be dealt with further on, in connection with analysis methods.

The bellows referred to may be those supplied by chemical apparatus specially for this work, and known as "Fletcher's aspirating bellows;" or they may be home-made ones, somewhat similar.

Turning now to the case in which it is desirable that the samples be collected very slowly, so as to represent more nearly the average composition, we find that a totally different mode of collection is necessary. A simple method would be to take a glass bottle, and fit a cork to it; through this cork bore two holes of suitable size for glass tubing. Through one of these holes pass a tube, so that one end nearly touches the bottom of the bottle, and bend the portion outside, so as to form a simple syphon. Through the other pass a tube, so that its lower end is just below the cork, and connect its upper end to the pipe communicating with the flue. If, now, water is drawn from the bottle by means of the syphon, gas will be caused to flow along the other tube into the bottle, and thus a sample of gas might be collected.

This, however, would not be a satisfactory arrangement if the sample had to be collected slowly, as the component gases of the chimney gas are more or less soluble in water; hence, if exposed to the surface of the water in the aspirating bottle they would be absorbed in different

proportions by the water, thus altering the composition of the sample. Again, if the sample were collected very slowly, any slight air leakage at the joints of the apparatus would have material influence on the sample.

The first-mentioned difficulty might be got over by using mercury instead of water as the fluid in the aspirating bottle. But to get over both difficulties a slight modification is necessary, and the apparatus thus arrived at is that

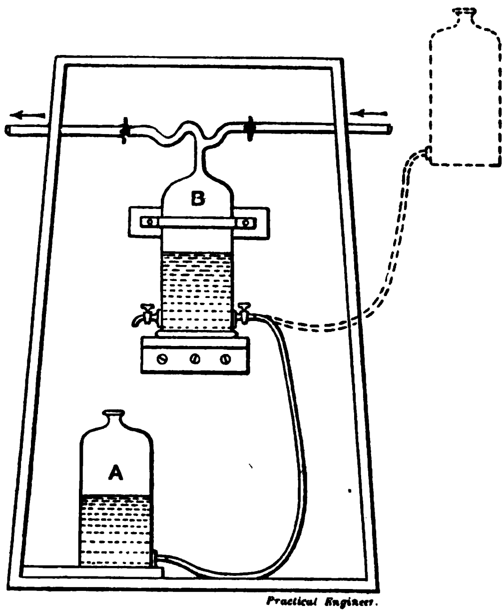


FIG. 2.—Stead's Gas Collector.

now very generally used. The aspirating bottle just described is retained, but is made of large capacity, often an ordinary vitriol carboy is used, and the syphon is arranged to draw a fairly large current of gas along the pipe leading from the flue. At some point between the carboy and the flue a small pipe is attached to the other pipe, and connected with a small aspirating bottle of 200 to 250 cubic centimetres capacity. The fluid in this small

aspirator preferably being mercury. This is then set so that the mercury runs out slowly, with the result that a very small quantity of gas is drawn from the comparatively large stream passing along the main pipe. The result of this arrangement is that a reliable sample may be obtained, whose collection extended over several hours, or even more.

A very convenient form of mercury aspirator has been designed by Mr. Stead, and is known as "Stead's gas collector." This is shown at fig. 2. When collecting, the bottle A is placed in the position shown immediately under the small tap in the bottle B, so that on opening the tap the mercury falls into A, and gas is drawn into the space vacated by the mercury. To expel the gas from the collecting bottle into the analysis apparatus or into the

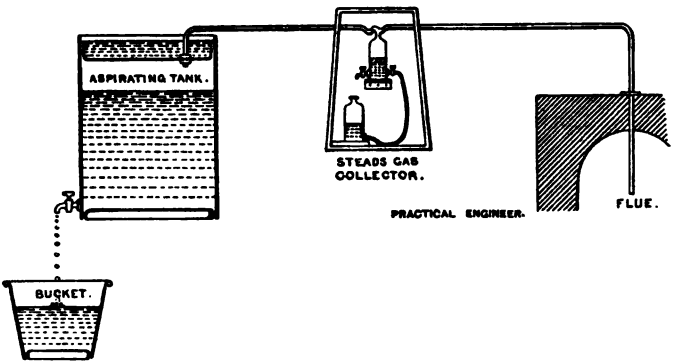


FIG. 3.—Sketch showing arrangement for collecting sample of flue gas.

atmosphere, the bottle A is raised to a position such as shown in dotted lines, and the tap on the right hand at the bottom is opened, causing the mercury to fall down the indiarubber tube and enter bottle B. This apparatus can easily be regulated so as to collect a sample in a minute or in several hours.

When fitting up the large bottle, or carboy, great care should be taken to make all the joints quite tight, and, if the cork used is of large diameter, it should be well coated with white lead or other suitable material, as large corks are frequently very porous. In consequence of difficulty in making some bottles that have been provided for aspirating purposes thoroughly air-tight, the writer advised the use of

a small cylindrical tank of galvanised iron instead of a glass bottle, the flat top of the tank being several inches below the upper edge of the cylindrical portion, so that the seam connecting the top to the sides of the tank and the connection of the pipe to the top might be covered with a few inches of water, and thus make air leakage impossible. Tanks of this description are used by the National Boiler and General Insurance Co. in their gas analyses, and have been found very satisfactory. Such a tank, also the general arrangement for the collection of flue-gas samples, are shown at fig. 3.

CHAPTER III.

HAVING collected a sample of the flue gases, the next point is to determine its composition by analysis. Gas analyses of an approximate character are not difficult to make, and it is not essential for the person making them to have had a special training as a chemist; it is, however, desirable that the general principles of chemistry be understood, and that considerable care be exercised in the various manipulations. For exact gas analyses complicated apparatus is necessary, together with considerable experience; hence such analyses are out of the scope of the ordinary engineer. It will therefore be understood that the various methods described in this chapter do not give strictly accurate results; although, except where otherwise stated, the results are probably sufficiently accurate for engineering purposes.

The following is a very simple method of determining the approximate composition of a sample of ordinary flue gas.

Take a tube sealed at one end, and graduated up to 50 or 100 cubic centimetres. Fill this with water, and then place the open end in a suitable vessel of water, and invert the tube; the closed end will then be at the top, the open end under water, and the tube full of water. Then bring the bellows or bottle containing the sample, and after discharging a little of the gas into the atmosphere, so as to remove all air from the rubber tube connected with the bellows, place the end of the rubber tube just in the lower end of the graduated glass tube filled with water (fig. 4*a*), and press some of the gas out of the bellows; this gas will rise through the water to the top of the graduated tube. Continue sending gas into the tube until it is nearly full;

then remove the bellows and get another glass tube about equal in length and diameter to the graduated tube, but open at both ends. Then connect an indiarubber tube about 2 ft. 6 in. long to this, and fill both with water, afterwards connecting the free end of the rubber tube to the open end of the graduated tube, taking care in making the connection that no air is admitted.

The next step is to place the two glass tubes vertically against each other, and adjust their heights until the water level in each is exactly alike. The volume of the gas in the graduated tube should then be read, as the gas will be under the atmospheric pressure exactly. Having determined the volume of gas taken, pour as much water as possible out

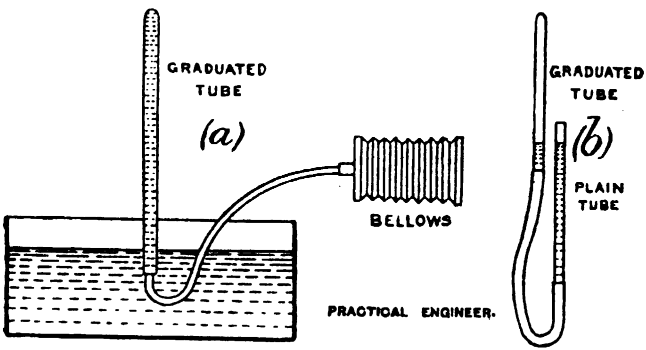


FIG. 4.—*a* shows method of passing sample of flue gas into tube. *b* shows graduated tube connected with plain tube by rubber tube, and tubes adjusted so that water in each is at the same level, and, consequently, gas sample at atmospheric pressure.

of the plain glass tube without either letting any gas escape or any air enter the graduated tube. Then, in the place of the water thus emptied, pour a strong solution of caustic potash until the tube is quite full; then cork the open end, and by raising and lowering the tubes alternately cause the gas to pass several times through the caustic potash solution. Then remove the cork, level the tubes, and read the volume of gas; cork up again, and pass the gas a few more times through the liquid, repeating this until no further change of volume takes place. The difference in the volume of the gas now and its original volume will give the amount of carbonic acid present in the sample. For

instance, if the original volume was 48·6 cubic centimetres (c.c.), and the volume after passing through the caustic soda solution, 43·2 c.c., the difference, 5·4 c.c., will be the volume of carbonic acid (CO_2) in 41·6 c.c. of flue gas, and will equal 11·1 per cent.

The explanation of the foregoing is that carbonic acid gas is very readily absorbed by caustic potash; hence, when the sample of flue gas is passed through caustic potash several times, all the carbonic acid present is absorbed.

Having determined the amount of carbonic acid, the next step is to determine the free oxygen. This is done by pouring out as much as possible of the caustic potash (just as the water was poured out), and in its place pouring a strong solution of pyrogallic acid in caustic potash, such a solution being a strong absorbent of oxygen.

By manipulating the tubes exactly as before, the gas is passed through this fresh solution until all the oxygen is absorbed from it, when the volume is carefully measured. We will suppose that the volume is now 39·1 c.c. Under these circumstances the volume of oxygen in 48·6 c.c. of the flue gas would be $43{\cdot}2 - 39{\cdot}1 = 4{\cdot}1$ c.c., and the percentage would be $\frac{4{\cdot}1}{48{\cdot}6} \times 100 = 8{\cdot}43$.

Ordinary boiler gases are composed, principally, of carbonic acid gas (CO_2), oxygen, nitrogen, and steam, with traces of sulphurous gases, and occasionally of carbonic oxide (CO).

The carbonic acid is the result of the combustion of the carbon in the coal with oxygen in the atmosphere, 1 part by volume of carbon combining with 2 parts of oxygen, forming carbonic acid, the resulting volume of which is equal to the volume of oxygen used. The oxygen thus used is drawn from the atmosphere, which is composed of, approximately, 1 part of oxygen to 4 parts of nitrogen. Thus, for every volume of oxygen drawn into the furnace. 4 volumes of nitrogen are also drawn, and, consequently, for every volume of carbonic acid shown by the analysis there will be approximately 4 volumes of nitrogen.

In boiler furnaces it is always found that the amount of air used is greatly in excess of the amount actually required to supply sufficient oxygen for the combustion of the coal; hence, when analysing flue gases, excess oxygen will always be detected. As such oxygen is accompanied by about four times its volume of nitrogen, the total nitrogen will be at least four times the volume of carbonic acid and oxygen

together. As a rule the nitrogen will be a little in excess of this amount, as some will be liberated from the oxygen used in the combustion of the hydrogen and sulphur in the coal, the hydrogen combining with oxygen to form steam (H_2O), and the sulphur with oxygen to form, probably, sulphur dioxide (SO_2).

The steam present in the flue gases condenses in the collecting apparatus, and therefore does not appear in the analysis. The sulphur dioxide also is not shown by the ordinary methods of analysis adopted, owing to it being very soluble in water, and, as its quantity is very small, it is hardly worth while adopting special methods of analysis for it, especially as it can be easily estimated from the coal analysis when desired.

Occasionally, when the supply of air to a furnace is limited, gas analysis will show a small quantity of carbonic oxide (carbon monoxide, CO)—that is, gas in which 1 part of carbon is combined with 1 part of oxygen, instead of with 2 parts, as is the case when the carbon is completely burned. The analysis apparatus just described would hardly be suitable for the determination of the last mentioned gas, owing to the small quantities in which it is present. This apparatus, although very simple, has several objections: first, the gas to be analysed passes through water when entering the graduated tube. This will tend to alter the composition of the sample, as the various gases of which it is composed are soluble in water to different extents. Risk of error from this source might be considerably reduced by passing a considerable quantity of the flue gas through the water before it is put in the graduated tube, so as to saturate it with the different gases.

Another source of error, and probably a very material one, is change of temperature of the gas during analysis, owing to warmth from the body and hands of the operator.

From what has been said, it will be seen that although the method described may be suitable for determining the approximate composition of flue gases, say, when it is desired to note if the coal is being burnt properly, it can hardly be considered sufficiently accurate if any important deductions or calculations have to be made from the analysis.

A great many arrangements have been devised by various chemists for technical analyses of flue gases [the expression "technical analyses" has been used to distinguish the class of analyses we are considering from exact analyses],

but those now most generally in use are known as Hempel's and Orsat's, the former being preferred for use in a laboratory, whilst the latter is generally preferred for outdoor work, owing to the compact form in which it is made.

CHAPTER IV.

Hempel's Apparatus.

Fig. 5 shows the burettes in which the gas is measured. A is a plain glass tube, whilst B is a carefully graduated tube connected to A by rubber tubing.

To use these, pour water into the burette A, and by raising this cause the water to flow into burette B until the latter is quite full; at this point the water should be visible in A, so as to ensure the indiarubber connecting tube being quite full. After this the upper end of the tube B is connected with the bottle or bellows containing the sample by means of the short indiarubber tube shown; the tube A is then lowered, causing the water to flow from B to A, and thus draw a portion of the gas sample into B. After a sufficient quantity (preferably 100 c.c.) has been taken in, the pinchcock should be placed on the rubber connection, and the volume of gas in B carefully read by means of the graduations. When reading the volume it is, of course, essential that the water in the two tubes be at exactly the same level, otherwise the gas will be extended or compressed according as the water in A is lower or higher than that in B. Also it is wise to make the reading at some definite time, say three minutes after the gas was drawn in, so that the water may drain down from the sides of the tube, and the same period of time should be allowed prior to each subsequent important reading, otherwise it is possible that more water will have drained down one time than another. Care should also be taken to see that the temperature remains constant during each analysis.

The manner of adjusting the burettes so that the water level may be the same in each is shown at fig. 6.

Having measured out a portion of the flue gas sample, the next step is to analyse it, remembering, of course, that we know it is made up of carbonic acid, oxygen, carbon monoxide, and nitrogen, any other gas being in such small quantity as to be negligible.

Starting first with the carbonic acid (CO_2), the amount of this is determined by passing the gas into an absorption pipette, as shown at fig. 7, the method of connecting being shown at fig. 8.

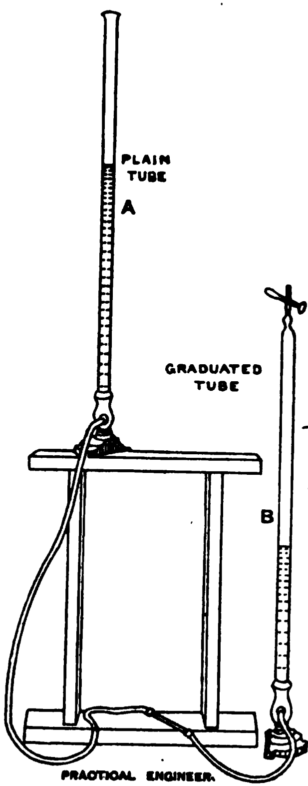


FIG. 5.—Hempel's Gas Burettes.

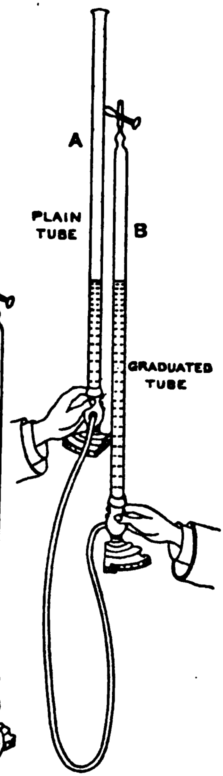


FIG. 6.—Method of levelling when measuring volume of gas.

The bulbs or pipettes A and B, fig. 7, contain strong caustic potash, which may be made by dissolving 1 part of commercial caustic potash with 2 parts of water. The tube *c* is of extremely fine bore, as should also be the tube

connecting it with the burette containing the gas to be analysed. Behind *c* is a white scale, and before connecting the burette the caustic potash in the bulbs should be drawn along tube *c* to some definite point on the scale; and on the completion of the absorption of the carbonic acid by the caustic potash, the latter should be taken to exactly the same point in tube *c* before the alteration in the volume of gas is read.

After connecting the burette B in the manner shown, the plain tube should be raised so that the water flowing into B will expel the gas into the caustic potash pipette. In doing this the caustic potash will be forced out of the lower bulb

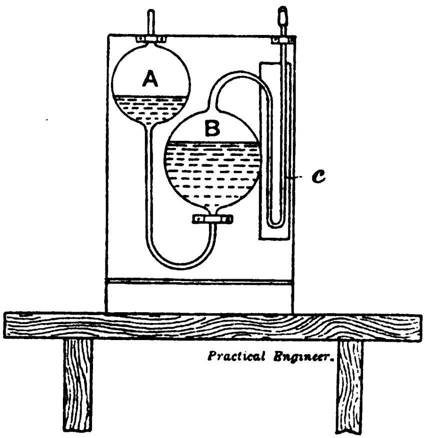


FIG. 7.—Hempel's simple absorption pipette.

into the upper one, which acts as a receiver. Leave the gas exposed to the solution for a while and shake well, then draw it back by lowering the open tube or burette; when the caustic potash is at exactly the same level in the capillary tube as at the start, close the pinchcock on the top of the graduated burette, and carefully measure the volume of the gas contained therein. The volume will be less than originally if the gas contained carbonic acid, as this gas is readily soluble in caustic potash. By repeating the process until no further reduction of volume occurs, the amount, and consequently the percentage, of carbonic acid will be determined.

The next step is to determine the oxygen. The style of pipette used for this is shown at fig. 9. It will be noted that there are here four bulbs. The two additional bulbs are not essential, but they tend to preserve the absorbing material, as, if only two were used, as for carbonic acid, the solution in the second bulb

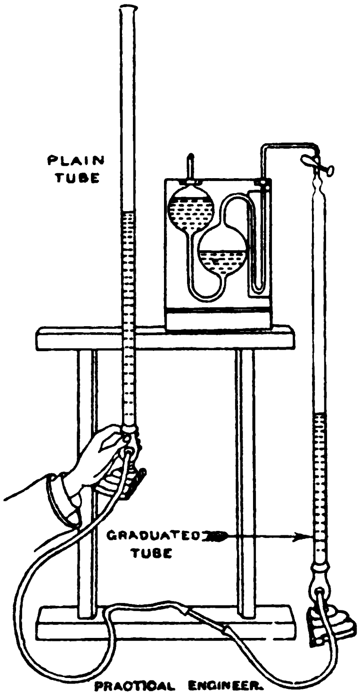


FIG. 8.—Method of using Hempel's apparatus.

would always be exposed to the atmosphere, from which it would be always absorbing oxygen; hence its value as an absorbent of the oxygen contained in gas samples would be greatly reduced. The two additional bulbs may therefore be considered as simply a means of enabling the oxygen absorbing fluid to be disconnected from the atmosphere.

The manipulation for the determination of oxygen is exactly the same as for carbonic acid.

The absorbent most generally used is a solution of pyrogallic acid in caustic potash. This may be prepared as follows: Dissolve 5 grains of pyrogallic acid in 15 c.c. of water; then dissolve 120 grains of caustic potash in 80 c.c. of water; then mix the two solutions together, and pour into the pipette. It should be noted that commercial caustic potash should be used, and not that purified with alcohol.

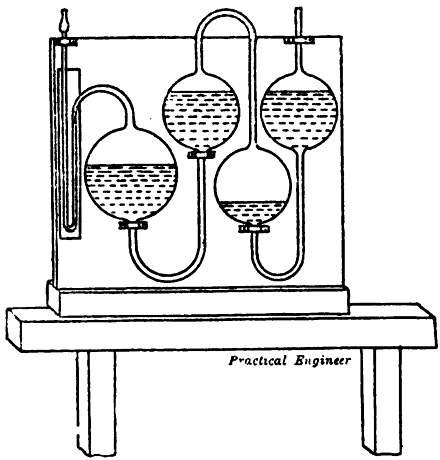


FIG. 9.—Hempel's double absorption pipette.

Instead of a solution of pyrallic acid and caustic potash (potassium pyrogallate), sticks of phosphorus in water may be used, and with them analyses can be more quickly performed. The principal objection to the use of phosphorus is the difficulty of making the sticks and transferring them to the pipettes.

For ordinary commercial gas analyses the determination of carbonic acid and oxygen is usually sufficient, but in connection with evaporative tests of boilers it is customary to also ascertain the quantity of carbon monoxide (CO) present. This is done in pipettes generally similar to those used in connection with carbonic acid, except that the

first bulb is often shaped so as to permit of solid matter (copper gauze) being inserted. (See fig. 10.) This style of pipette is also often used for carbonic acid.

The absorbent used for carbon monoxide may be an ammoniacal solution or a hydrochloric acid solution of cuprous chloride; the manipulation of the burettes being exactly the same as before.

The ammoniacal solution of cuprous chloride may be prepared as follows:—

Digest metallic copper and copper oxide (CuO) with strong hydrochloric acid (HCl) until solution is colourless. Pour solution into water; this causes the cuprous chloride

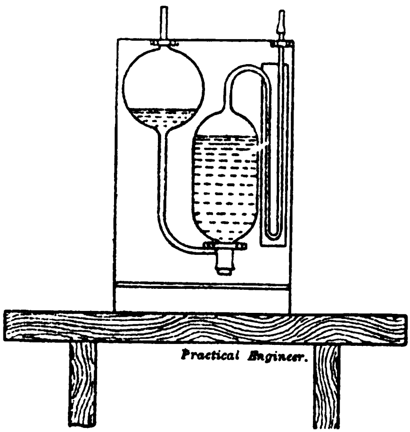


FIG. 10.—Hempel's pipette for solid substances.

to separate out as a white powder. Wash this powder quickly by decantation; then add just enough ammonia to dissolve the powder. Keep in a well-stoppered bottle.

The acid cuprous chloride may be purchased ready made as a salt, and only requires dissolving, or it may be made as follows (Winkler): Mix 86 grains of copper scale with 17 grains of copper powder, prepared by reducing copper oxide with hydrogen; mix carefully with 1,086 grains of hydrochloric acid of 1·124 sp. gr. Pour in bottle, and place spirals of copper in bottle. The solution will be dark at first, but will become colourless after a while. The absorbing power of the acid solution is hardly so great as

of the ammoniacal solution, but either solution may be used; it seems, however, probable that the ammonia solution is slightly preferable.

The other form of apparatus to be considered is that commonly known as Orsat's. As has been already mentioned, this apparatus is the one most generally used for outdoor work, owing to its compactness. (See fig. 11.)

A is a three-way tap.

B is a graduated pipette of 100 c.c. capacity.

B_1 is a bottle corresponding to the plain tube in Hempel's arrangement.

C is a jacket round the graduated tubes, to lessen the risk of the gas in the tube being altered in temperature.

D are simple stop taps.

E, F, and G are vessels containing the various absorbents.

E_1, F_1, and G_1 are reservoirs into which the absorbents may flow.

To introduce the gas into the analysis apparatus, first fill the pipette B with water by raising the bottle B_1. During this time the three-way cock A must be open to the atmosphere. Then turn the cock so that gas may flow from the sample collected, along the horizontal tube into the pipette B, when the bottle is lowered so as to cause the water to flow from it. Draw in a definite quantity of gas, preferably 100 c.c.; then shut tap A [when measuring the volume of the gas the height of the bottle must, of course, be adjusted until the water in it is exactly at the same level as the water in the graduated tube]. After this, by raising and lowering the bottle and opening taps D, one at a time, the gas may be passed in the flasks or bottles E, F, and G containing the absorbents, exactly as in the case of Hempel's apparatus, the only difference being in the shape of the bottles. It is usual to put a number of pieces of glass tube in each bottle, so that the gas may be exposed to a greater surface of the absorbing liquid, owing to the latter adhering to the tubes.

It might be well to add here that the determination of the oxygen alone would be very useful in showing how the coal is being burnt, and for this purpose an Orsat's apparatus, with only one pipette for absorbent, might be obtained in a

very portable form. For the determination of oxygen alone phosphorus must be used, instead of the potassium pyrogallate solution, as the latter would absorb the carbonic acid also, carbonic acid being so readily absorbed by caustic potash. For this reason it is essential that the carbonic

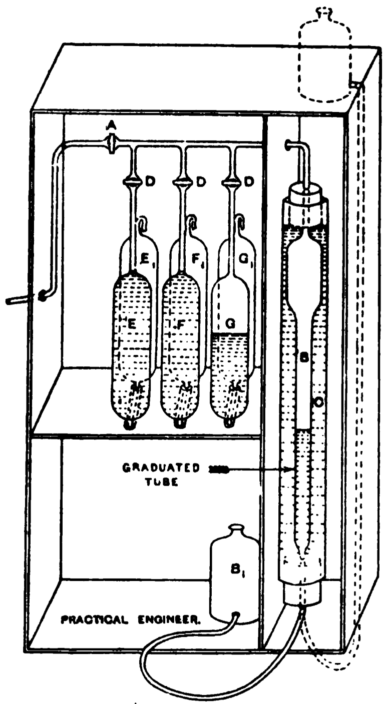


FIG. 11.—Orsat's apparatus for gas analysis.

acid be determined before the oxygen whenever potassium pyrogallate is used.

In well-managed boiler furnaces the oxygen will be about 8 per cent, and the carbonic acid 11 to 11½ per cent, but as a rule the excess of oxygen will be greater than this. A more common analysis would be 7 to 8 per cent carbonic acid, 11

to 12 per cent oxygen. The oxygen and carbonic acid together should form about 19½ per cent of the total sample. Hence, roughly speaking, the oxygen will equal 19·5 − carbonic acid, and the carbonic acid will equal 19·5 − oxygen. This rule may be used as a rough check on an analysis.

CHAPTER V.

Testing the Dryness of the Steam.

The accurate determination of the dryness of the steam presents various difficulties, and, in consequence, many tests of engines and boilers are made without any effort being made to determine the condition of the steam; whilst at other times the determinations made show such irregular results as to be quite unreliable.

Numerous methods have been suggested for the testing of steam, but only four appear to have been used commercially to any material extent, and these are as follow: (1) The calorimetric method of mixture—*i.e.*, the method of mixing a known weight of steam with water, and by the resulting temperature determining the dryness factor. (2) The method of separating the water from the steam by means of a separator. (3) The chemical method—*i.e.*, by condensing a portion of the steam and then determining by analytical methods the percentage of some kind of salt in it, and comparing this with the percentage of a similar salt in a sample of water drawn from the boiler. (4) The wire-drawing method—*i.e.*, the method of blowing steam from one vessel through a very fine opening into another, the pressure in the latter being usually that of the atmosphere, and noting the rise of temperature.

One way in which the calorimetric method (1) is very commonly carried out is as follows: An ordinary oil barrel is taken and carefully weighed; afterwards water is poured into it and weighed, and its temperature noted; after this a small pipe leading from the main steam pipe is carried into the water, and a quantity of steam blown through it. This steam is condensed by the water in the barrel, and the increase of weight and temperature of the water in the barrel are carefully recorded.

From the data thus obtained the dryness of the steam may be calculated. As an example of the calculation we will take the following case:

Weight of water in barrel at start, say 120 lb.
Temperature of water in barrel at start, say 70 deg. Fah.
Weight of water in barrel at finish, say 124 lb.
Temperature of water in barrel at finish, say 105 deg. Fah.
Pressure of steam blown in, say 65 lb. per square inch.

The total number of units of heat in the water in barrel at start $= 120 \times 70 = 8{,}400$ units.

Total number at finish $= 124$ (weight of water at finish) $\times$ $105 = 13{,}020$ units.

$\therefore$ increase of heat in barrel $= 13{,}020 - 8{,}400 = 4{,}620$ units.

$\therefore$ this = number of units of heat given up by the 4 lb. of steam blown into the barrel.

$\therefore$ units per pound $= \frac{4{,}620}{4} = 1{,}155$.

On reference to saturated steam tables we find that 1 lb. of steam at 65 lb. pressure above atmosphere contains 1,208·5 units of heat; whilst 1 lb. of water at the temperature corresponding to this pressure contains 312 units.

Let x = the weight of water mixed with each pound of steam in the condition taken, then the weight of dry steam would be $1 - x$, and the total heat of the water would be $x \times 312$, and of the steam $(1 - x) \times 1{,}208{\cdot}5$, and the sum of these two must equal 1,155.

$$\therefore\ 312\,x + 1{,}208{\cdot}5\,(1 - x) = 1{,}155\,;$$

$$x = {\cdot}056\,;$$

$\therefore$ moisture $= {\cdot}056$ in one pound, or 5·6 per cent.

This method certainly is simple, but unfortunately, with the methods of weighing usually available, is not reliable. Again, a slight inaccuracy in measuring the temperature causes material error; for instance, if the temperature in the example just taken had been read as 104½ instead of 105 deg. Fah. (and such an error might easily be made; in fact, it would be difficult to avoid), the moisture percentage would have been 6·9 per cent instead of 5·6 per cent, a very material difference.

The late Mr. Willans, of Messrs. Willans and Robinson, used this calorimetric method in the engine tests he made, and which were fully described in two papers read before

the Institute of Civil Engineers, but in his case he used a large specially made tank, together with very finely divided thermometers and very sensitive weighing apparatus, and by these means probably attained great accuracy, but it would be impracticable to adopt such arrangements in most tests.

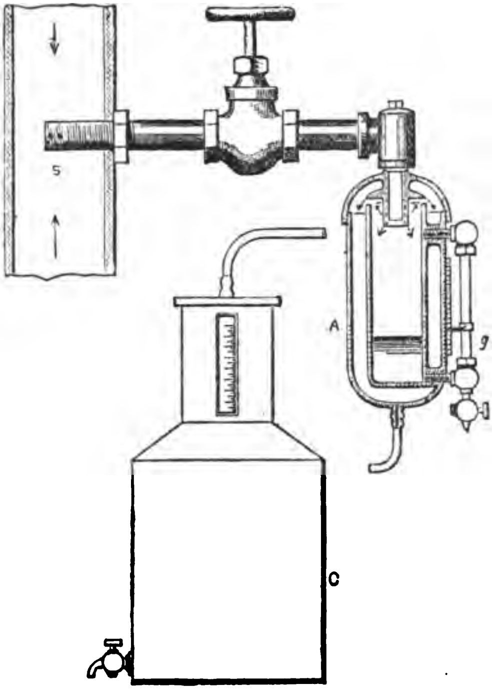


Fig. 12.—Carpenter's separating calorimeter.

On the whole the writer is of opinion that the barrel method is hardly sufficiently reliable as usually carried out for the figures obtained from it to be used with any confidence.

The separation method (2) consists of passing either the whole or merely a sample of the steam through some

arrangement which tends to cause the water to be precipitated from the steam. In connection with this method the general experience seems to be that when dealing with small quantities of steam the water can be almost entirely separated, but that this cannot conveniently be done when dealing with large quantities of steam. Hence in the separation method it is better to use a small special separator and pass a sample of steam through this, rather than rely on the separation effected by a separator through which the whole steam supply passes, although in all cases the last-mentioned amount should be observed. A very convenient form of separator has been designed by Professor Carpenter, of Cornell University, and is shown at fig. 12.

The pipe from the lower end is led into a barrel of water, so as to cause condensation of the steam issuing from it. After a reasonable quantity of water has collected in the separator, the increase of weight of the water in the barrel is noted, and thus the percentage of moisture in the steam is determined. From numerous experiments made by a special committee of the British Association, it was found that the results obtained with this separator were generally very satisfactory.

The chemical method (3). In this, a known amount of some salt, generally common salt (NaCl) is introduced into the boiler. Then an estimate of the dryness of the steam is made either by determining the decrease of the saltness of the water in the boiler, owing to salt being carried away with the priming water, or condensing a sample of the steam, and ascertaining by analysis the quantity of salt present, and by deduction from this the quantity of water mixed with the steam. Of course, analysis of a sample of the water in the boiler should be made at the same time, so that its exact degree of saltness may be known.

The following method of analysis, devised by Mr. C. J. Wilson, of University College, London, is recommended. To 1 part of the salt boiler water add 100 parts of pure distilled water; then mix with this a small quantity of concentrated solution of yellow chromate of potash. Then a $\frac{1}{10}$ per cent solution of nitrate of silver is slowly added. With each drop the solution turns locally red, but this disappears on shaking. When all the salt has been acted upon, the whole fluid changes colour from pale yellow to orange. The quantity of nitric solution is noted, and then the experiment is repeated on a condensed sample of the steam undiluted with water. The ratio of the quantities of nitrate in each case expresses the priming in per cent.

The principal objections to the chemical methods are: (1) Great exactness is required in the analyses; (2) the saltness of the water in the boiler is very probably not uniform throughout, in which case the results will, of course, be unreliable.

The wire-drawing method (4) appears to have been first proposed by Professor Peabody, of the Massachusetts Institute of Technology, and is based on the fact that if dry saturated steam is expanded from a higher to a lower pressure without doing external work it becomes superheated, owing to the total heat in the steam remaining the

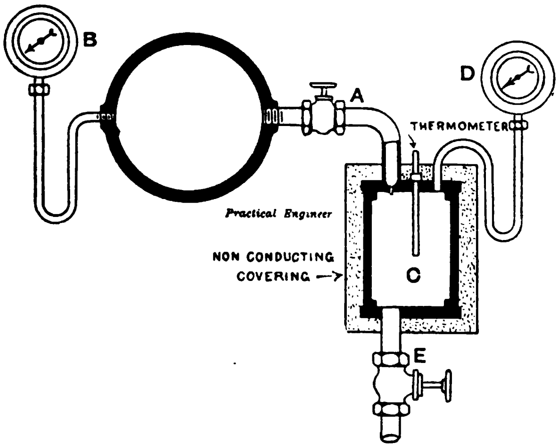


FIG. 13.—Peabody's wire-drawing calorimeter.

same, whilst the total heat of saturated steam falls with the pressure; hence the surplus heat causes superheating.

The apparatus proposed by Professor Peabody is generally as per fig. 13. A is a small pipe leading from the steam pipe to the small chamber or cylinder C; B is a pressure gauge connected to the main steam pipe, and D is a similar gauge connected with the chamber C. The steam is carried away from the chamber C by the small pipe E. All parts of the apparatus should be well covered with some non-conducting material to prevent loss by radiation. It will be noted that the small pipe A is much reduced in area at the point of connection to the chamber; this is to reduce

the amount of steam passing through the apparatus, otherwise the chamber C would have to be unduly large. On opening the valve or tap on A, steam rushes through the small opening and expands into C, passing from here to the atmosphere along the pipe E, which should be of fairly large diameter. The temperature after expansion and the pressure before should be carefully noted, also the pressure in C should be determined, although, as regards the latter, it will usually be satisfactory to open the tap on E wide, and only have atmospheric pressure in C. From the data thus obtained the dryness factor for the steam can be calculated in the manner to be described.

Another simple apparatus, based on the wire-drawing method, and designed by Mr. Barrus, is shown at fig. 14. In this the steam enters the chamber A, and passes through a small hole, about $\frac{1}{16}$ in. diameter, into B, the latter having a free outlet to the atmosphere. The temperature in each chamber is noted by means of the thermometers shown.

The steam dryness may be calculated from these temperatures as follows :—

Let t_1 = the temperature of the steam in A.
t_2 = the temperature of the steam in B.
t_3 = the temperature of saturated steam at the presrure in B.
x = the dryness factor of the steam.
L_1 = the latent heat of steam at temperature t_1.
L_3 = the latent heat of steam at temperature t_3.

The total heat in each pound of steam in B will be equal to the total heat of each pound of steam in A, as there is practically no external work done by the steam, and therefore no loss of heat.

The total heat per pound of steam in

$$A = (1 - x)\,t_1 + (t_1 + L_1)\,x$$
$$= t_1 + x\,L_1.$$

The total heat per pound of steam in B, after superheating by wire-drawing, $= t_3 + L_3 + \cdot48\,(t_2 - t_3)$; ·48 being the specific heat of dry steam.

$$\therefore\; t_1 + x\,L_1 = t_3 + L_3 + \cdot48\,(t_2 - t_3)$$

$$x = \frac{t_3 + L_3 + \cdot48\,(t_2 - t_3) - t_1}{L_1}$$

But t_3 is temperature of steam at a pressure very slightly above the atmospheric, and may be taken as 214 deg. Fah.; $\therefore$ L_3 will be nearly 966.

L_1 may be calculated from the formula—

Latent heat = 1116 + 0·7 × temperature.

$$\therefore\ x = \frac{214 + 966 + \cdot 48\,(t_2 - t_3) - t_1}{1116 + 0{\cdot}7\,t_1}$$

$$= \frac{1077 + \cdot 48\,t_2 - t_1}{1116 + 0{\cdot}7\,t_1}$$

From this formula it will be seen that with this method the only observations to be made to enable the steam

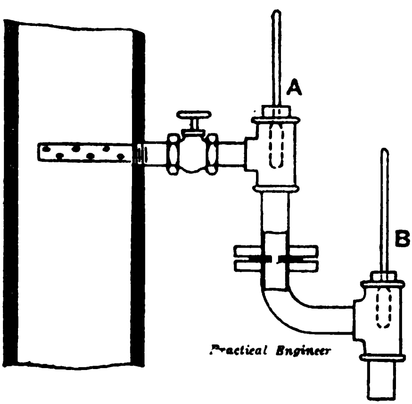


FIG. 14.—Barrus' calorimeter.

dryness to be ascertained are the temperatures in the chambers A and B. No weighing being necessary, the apparatus can be used continuously if desired, which would appear to be a great advantage, as observations as to the character of the steam under various conditions of the fires may then be made.

Professor Unwin mentions that the committee previously mentioned made a number of tests of the Barrus wire-drawing calorimeter, and found it reliable and satisfactory. It must, however, be noted that the formula given is based on the assumption that the steam in B is superheated. If

the moisture present with the steam in A is so great that, after wire-drawing, it is not all evaporated and superheated at least 6 deg., the apparatus should not be used in the form shown. In such a case, probably the best way would be to first bring the steam through a small separator and afterwards through the wire-drawing instrument. Of course, in such circumstances the steam would have to be condensed and weighed, so that the water collected in the separator might be compared with it.

Turning from the apparatus for the determination of steam dryness to the consideration of the drawing of the sample of the steam, we find that the point from which the steam should be taken varies according to the object of the test. If the test is one of an engine simply, the steam should be taken from as close to the engine as possible, so that there may be no question as to the after effect of exposed steam pipes, or of any form of steam separator. For similar reasons the steam should be taken from close to the junction valve if the test is of the boiler alone.

The small pipe connected with the large steam pipe should always project inwards a little way, and should preferably be perforated with small holes and have the end blocked up, so that the steam may be drawn from various parts of the stream moving along the larger pipe, and thus be fairly representative of the whole bulk. It is also preferable to draw the steam from a vertical pipe, as with horizontal pipes there is a tendency for the water to gravitate to the bottom of the pipe.

(*i*) Height of Barometer and State of Weather.

It is desirable in most tests that the atmospheric pressure be noted at intervals, as this pressure, added to that shown by the various gauges, gives the true absolute pressures at various parts of the apparatus. A knowledge of the various fluctuations in the pressure of the atmosphere, and of the weather conditions during a test, also is sometimes of assistance in the investigation of the reasons for conflicting results in different tests.

Economiser.

Leaving the boiler, we come next to the economiser, which we will assume to be of the ordinary Green's type. Here we find that the additional observations to be made are not numerous, and as the instruments required for them are

only such as are used in connection with the similar observations for the boiler, it will be sufficient if the observations are enumerated simply. They are—

(*a*) Temperature of feed water entering and leaving economiser.
(*b*) Temperature of flue gases entering and leaving economiser.
(*c*) Temperature of air over economiser.
(*d*) Pressure at inlet and outlet ends of economiser.
(*e*) Collection of samples of flue gases entering and leaving economiser.
(*f*) Measurement of leakage at safety valves or any other fittings.

It may here be noted that the observations just mentioned do not include all those necessary to determine the efficiency of an economiser, but are to be taken in conjunction with those relating to the boilers.

The methods of measuring temperatures and of collecting and analysing samples of the flue gases have already been fully explained in connection with the boiler, and therefore do not require further mention.

ENGINES.

The number of observations to be made in the test of an engine depends, of course, upon the object of the test and the type of the engine; hence it must not be considered that those given in the following list cover everything, but observations must be added or omitted to suit the circumstances of each particular case.

(*a*) Indication.
(*b*) Counting of speed.
(*c*) Reading of various pressure and vacuum gauges.
(*d*) Temperature of injection and hot-well water.
(*e*) Measurement of water leaving all cylinder and jacket drains and leaking at any joints or glands.
(*f*) Determination of frictional resistances.

(*a*) INDICATION.

A sufficient number of indicators should be supplied, so that both ends of each cylinder can be indicated simultaneously, the indicator pencil being allowed to travel

over the paper several times. The object of indicating all the ends at once is to avoid error owing to change of load in the period which would necessarily elapse if the indicators had to be removed from one position to another.

It is hardly necessary to mention that great care should be taken to have accurate springs in the indicators, or, what is equivalent, to know the exact scale of each spring. As stated in one of the chapters on indicating, the best method of testing the springs would appear to be on a steam boiler against a standard pressure gauge, each spring being in its own indicator.

If, on thus testing, the spring is not found to be of exactly one scale throughout its range—*i.e.*, of uniform elasticity—the best way is to construct a special scale for each spring. This, however, prevents a planimeter being used to measure up the mean pressure; hence each diagram must be divided into a number of equal parts, and measured with its own scale. The greater the number of divisions into which the diagram is divided, the greater will be the accuracy of the result.

After taking a diagram, it is always wise to examine it closely and measure the initial and back pressures, as sometimes slight sticking occurs, and causes inaccurate diagrams, and, of course, is very serious in a test if not detected.

(*b*) Counting of Speed.

For counting the speed some form of counter should be used, as it is not sufficiently accurate to simply count the speed over a minute, several times during the test. The counter should either be started from zero or its reading taken immediately at the commencement of the test, and the reading should be taken exactly at the finish, as well as, say, every ten minutes during the test.

When steadiness in speed is an important factor, a Moscrop recorder is a desirable adjunct, as this shows in a very convenient manner the fluctuations in speed.

(*c*) Reading of Pressure and Vacuum Gauges.

It is wise to note the various gauges about every ten minutes during the test.

Where steam pipes are overhead, it will often happen that the small pipes connecting the gauges with them are filled with water, which water exerts a pressure on the gauge, in addition to the steam pressure, and must, therefore, be

allowed for in determining the true pressure. It is hardly necessary to say that steps should be taken to check each gauge prior to the test.

(*d*) Temperature of Injection and Hot-well Water.

These temperatures should be noted regularly throughout the test. Often the construction of the hot well, or the splashing of the water therein, will prevent the temperature being taken directly in the hot well, in which case it should be taken at the outlet from the water pipes, unless these are of such length that the temperature falls perceptibly between the hot well and the outlet.

The temperature of the injection water will usually have to be taken at the point of the reservoir or other storage from which it is drawn.

Where surface condensers are used, the temperature of the condensing water should be noted, both before entering the condenser and after leaving it.

The thermometers used need only be of the ordinary type, ranging to about 150 deg. Fah.; but it is desirable that the mercury bulb dip in a small well formed in the thermometer casing, so as to hold some of the water whose temperature is being noted, as by this means the temperature of the thermometer itself does not fall so rapidly when it is lifted out of the water for reading.

(*e*) Measurement of Drain Water from Cylinders, &c.

The water from the steam traps connected with the cylinder and jacket drains should be measured in suitable vessels, as should also all water leaking from glands, joints, or other parts.

It will sometimes happen that a steam trap will discharge steam instead of water only; if so, the waste pipe should be led into a covered tub containing a known weight of water, so as to condense the steam, and enable the increase of weight to be noted. If the water in the tub gets too near boiling point, a few bucketsful should be taken out and substituted by an equal amount of cold water. Instead of the tub arrangement, the pipe from the jacket may be connected with a suitable receiver, and the amount of water which collects in the receiver during the test be measured by direct weighing.

(*f*) Determination of Frictional Resistances.

In the case of small engines, it is possible to determine the frictional resistances by applying some form of brake, thus ascertaining directly the actual H.P. developed by the

engine: this quantity, deducted from the I.H.P., will give the frictional resistance. With larger engines, the testing by brake becomes impracticable, and the frictional resistances are determined by means of diagrams taken with all machinery disconnected.

A very simple form of brake, and one which is very much used in the testing of gas engines up to about 60 I.H.P., consists simply of either a leather strap, or an iron strap with wood blocks, passed round the flywheel (see fig. 15). The free end A is connected to a spring balance, whilst weights are attached to the other end.

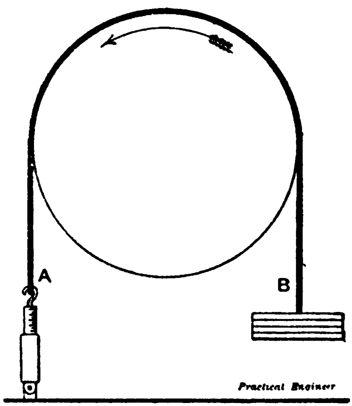


FIG. 15.—Friction Brake.

Let W = weight hung on strap,
P = pull of spring balance;
Then $W - P$ = frictional resistance on surface of wheel in lbs.;
Let D = diameter in feet,
N = number of revolutions per minute;
Then frictional resistance overcome in 1 revolution

$$= (W - P) D \times 3{\cdot}14;$$

Then frictional resistance overcome in 1 minute

$$= (W - P) D \times 3{\cdot}14 \times N;$$

$$\therefore \text{H.P.} = \frac{(W - P) D \times 3{\cdot}14 \times N}{33{,}000}$$

Various kinds of brake are used, but the mode of calculation just used will apply to most of them.

CHAPTER VI.

HAVING now generally outlined the different observations to be made in an efficiency test of a steam plant, consisting of engine, boilers, and economisers, the next step is to see how these observations can be made use of.

Starting with the coal, we find that the analysis made of this enables us to calculate its heating value, also to calculate the weight of flue gases discharged.

HEATING OR CALORIFIC VALUE OF THE COAL.

As we have already seen, coal is usually composed of carbon, hydrogen, sulphur, oxygen, nitrogen, and ash, in varying quantities. Of these, the three first mentioned are the givers of the heat, by means of their combustion with oxygen from the atmosphere. The carbon combines with oxygen, and forms either carbonic acid gas (CO_2) or carbonic oxide (CO). The hydrogen combines with oxygen, and forms steam (H_2O); whilst the sulphur also combines with oxygen, and forms an oxide of sulphur. From experiments it has been found that the burning of 1 lb. of carbon to carbonic acid gives out 14,544 units of heat. It may happen, however, that the carbon combines with only one part of oxygen, instead of two, forming carbonic oxide instead of carbonic acid, in which case the combustion of 1 lb. of coal gives out only 4,451 units of heat.

HEAT EVOLVED DURING COMBUSTION.

	Units.
1 lb. of carbon burning to carbonic acid evolves...	14,544
1 lb. of carbon burning to carbonic oxide evolves..	4,451
1 lb. of hydrogen burning to steam evolves.........	53,337
1 lb. of sulphur burning to sulphurous gas evolves	4,014

As mentioned in connection with coal analysis, the oxygen in coal is not determined by direct analysis, and it is generally considered as being in combination with the hydrogen in some way; hence the hydrogen available for heating purposes in a coal is the amount shown by the analysis, minus the amount in combination with the oxygen, which will be one-eighth of the amount of the latter, for the reasons which will be explained shortly.

Experiments have shown that the heating power of a compound body is approximately equal to the sum of the heating powers of its component parts. This being the

case, it becomes possible from the analysis to calculate fairly closely the heating power of a coal. It must, however, be borne in mind that this heating value is not a strictly accurate quantity, although probably fairly close to the true heating value; hence, in calculations based on the heating value, it would not be wise to carry the figures into decimal places.

One cause for the approximate result lies in the variability of the heating value of carbon, according to the state in which the carbon exists. For instance—

	Units.
Heating value of 1 lb. of carbon as wood charcoal =	14,544
" " " from gas retorts =	14,485
" " " as natural graphite =	14,035

Suppose now that the coal has the following composition:

Carbon	=	·80 lb.
Hydrogen	=	·04 lb.
Oxygen	=	·04 lb.
Nitrogen	=	·06 lb.
Sulphur	=	·02 lb.
Ash	=	·04 lb.

Then the free hydrogen would be—

$$\cdot 04 \text{ lb.} - \frac{\cdot 04}{8} = \cdot 035 \text{ lb.}$$

Therefore heating valve of coal becomes—

		Units.
Carbon	14,544 × ·80 =	11,635
Hydrogen	53,337 × ·035 =	1,867
Sulphur	4,014 × ·02 =	80
		13,582

From the foregoing it will be seen that it is customary to calculate the heating value of a compound body by first determining by analysis its composition, and then calculating the heat value of each of its combustible components.

CARBON VALUE.

The carbon value—that is, the value of the coal compared with an equal weight of carbon—would be

$$\frac{13{,}582}{14{,}544} = \cdot 93.$$

A direct test of the calorific value, by means of a calorimeter, is generally useful as a check on the calculated value, but with the calorimeters in common use the resulting number is only approximate, and it must be remembered that with most calorimeters the steam formed by the combustion of the hydrogen in the coal is condensed to water; hence more heat is obtained from it than if it passed away in a gaseous state, as in flue gases.

Quantity of Flue Gases.

It has already been explained that the combustion of carbon, also of hydrogen and sulphur, is the chemical combination of these elements with oxygen. Thus—

$$C + 2\,O = CO_2 \text{ (carbonic acid)}$$

$$2H + O = H_2O \text{ (steam)}$$

$$S + 2O = SO_2 \text{ (sulphur di-oxide).}$$

In the case of carbon this would read, in every-day language, that 1 part by volume of carbon combines with 2 parts by volume of oxygen, and forms carbonic acid. Introducing the atomic weights, we find that 12 parts by weight of carbon combine with 32 (2 × 16) parts of oxygen, or 1 part by weight with $\frac{32}{12} = 2{\cdot}66$; that is,

$$1 \text{ lb. of carbon} + 2{\cdot}66 \text{ lb. } O = 3{\cdot}66 \text{ lb. } CO_2.$$

Therefore, with complete combustion, 3·66 lb. of carbonic acid is formed for every pound of carbon;

$$\therefore \text{ weight of } CO_2 \text{ per pound of coal} = 3{\cdot}66\,x,$$

where x = weight of carbon in 1 lb. of coal.

Similarly, $2\,H + O = H_2O$

atomic weights $(2 \times 1) + 16 = 18$

$$1 \text{ lb. of } H + 8 \text{ lb. of } O = 9 \text{ lb. } H_2O;$$

that is, 9 lb. of steam is formed per pound of oxygen.

$$S + 2O = SO_2$$

$$32 + (2 \times 16) = 64$$

$$1 \text{ lb.} + 1 \text{ lb.} = 2 \text{ lb.};$$

that is, 2 lb. of SO_2 is formed per pound of sulphur.

If the combustion of the carbon is not complete, and carbonic oxide (CO) is formed instead of carbonic acid (CO_2), the weight of gas will be as follows—

$$C + O = CO$$

atomic weights $\quad 12 + 16 = 28$

$$1 \text{ lb.} + 1{\cdot}33 \text{ lb.} = 2{\cdot}33 \text{ lb.};$$

that is, 2·33 lb. of carbonic oxide is formed per pound of carbon so burned.

Taking the coal as being of the composition considered in the calculations of the calorific power, we can now calculate the weight of the products of combustion, also of the total amount of oxygen required.

·80 lb. of carbon would give $\cdot80 \times 3{\cdot}66 = 2{\cdot}928$ lb. carbonic acid
·04 lb. of hydrogen would give $\cdot04 \times 9 = \cdot36$ lb. steam
·02 lb. of sulphur would give $\cdot02 \times 2 = \cdot04$ lb. sulphurous gas

3·328 lb.

In the above it has been assumed that the oxygen has been supplied pure, but in ordinary combustion the oxygen would be taken from the atmosphere, of which it forms about 20·9 per cent, or, approximately, one-fifth, the remainder being nitrogen. The weight of nitrogen may be calculated by multiplying the weight of oxygen by 4, or, more exactly, 3·78. Then—

Oxygen used by carbon $= \cdot80 \times 2{\cdot}66 = 2{\cdot}128$ lb.
Oxygen used by hydrogen $= \cdot04 \times 8 = \cdot32$ lb.
Oxygen used by sulphur $= \cdot02 \times 1 = \cdot02$ lb.

2·468 lb.

$$\therefore \text{ nitrogen} = 2{\cdot}468 \times 3{\cdot}78,$$

or, approximately, $\quad 2{\cdot}5 \times 4 = 10$ lb.

This, added to 3·3, gives the total weight of the products of combustion per pound of coal without excess of air; it is not, however, uncommon for an excess of air equal to the amount of air actually required to be used. The exact quantity is shown by the analysis made of the gases leaving the boiler.

Suppose that the analysis of the gases leaving the boiler was as follows :—

Carbonic acid	10 per cent
Carbonic oxide	nil
Oxygen......................................	9 per cent
Nitrogen (by difference)	81 per cent
	100 per cent.

A cubic foot of carbonic acid at a temperature of 32 deg. Fah. and a pressure of 29·9 in. of mercury weighs 0·122 lb. Hence at this temperature and pressure, 3·66 lb. of carbonic acid (the amount produced by the combustion of 1 lb. of carbon) will be $\frac{3{\cdot}66}{{\cdot}122} = 30$ cubic feet. Therefore, 30 multiplied by the weight of carbon burnt per pound of coal will give the volume of carbonic acid per pound of coal for a temperature of 32 deg. Fah., and a barometric pressure of 29·9 in. of mercury.

In our case we will suppose that of the ·8 lb. carbon present in the coal ·7 lb. is actually burned, the remaining ·1 lb. being lost with the ashes. Then the volume of CO_2 will be

$$30 \times {\cdot}7 = 21 \text{ cubic feet.}$$

From the gas analysis it will be seen that this 21 cubic feet represents 10 per cent of the total volume of flue gas shown by analysis ; hence total volume = 21 × 10 = 210 ;

$$\begin{aligned} \therefore \text{volume of } CO_2 &= 210 \times {\cdot}1 = 21 \\ \text{,, \quad ,, \quad } O &= 210 \times {\cdot}09 = 18{\cdot}9 \\ \text{,, \quad ,, \quad } N &= 210 \times {\cdot}81 = 170{\cdot}1 \\ & \overline{\quad 210 \quad} \end{aligned}$$

As the oxygen is present with 3·78 its volume of nitrogen, perhaps the better way would be to write the above result thus :

$$\begin{aligned} \text{Volume of } CO_2 &= 21 \text{ cubic feet.} \\ \text{,, \quad ,, \quad air} &= 90{\cdot}4 \text{ ,, \quad ,,} \\ \text{,, \quad ,, \quad N} &= 98{\cdot}6 \text{ ,, \quad ,,} \\ & \overline{\quad 210 \quad} \end{aligned}$$

The weight of a cubic foot of CO_2 at 32 deg. Fah. and 29·9 in. of mercury pressure = ·122 lb.

The weight of a cubic foot of air at 32 deg. Fah. and 29·9 in. of mercury pressure = ·081 lb.
The weight of a cubic foot of N at 32 deg. Fah. and 29·9 in. of mercury pressure = ·078 lb.

Hence weights of flue gases become—

$$
\begin{aligned}
CO_2 &= 21 \times {\cdot}122 = 2{\cdot}56 \text{ lb.} \\
\text{Air} &= 90{\cdot}4 \times {\cdot}081 = 7{\cdot}32 \text{ lb.} \\
N &= 98{\cdot}6 \times {\cdot}078 = 7{\cdot}49 \text{ lb.} \\
& \phantom{= 98{\cdot}6 \times {\cdot}078 =\ } \overline{17{\cdot}37}
\end{aligned}
$$

CHAPTER VII.

As previously mentioned, the steam resulting from the combustion of the hydrogen in the coal, and from the evaporation of such moisture as may occur in the coal, is not shown by analysis, also the sulphur gases are not shown; hence these must be calculated for separately.

$$\cdot035 \text{ lb. H would give } \cdot035 \times 9 = \cdot225 \text{ lb. of steam.}$$

If, in addition, there was 2 per cent free moisture in the coal, the total weight of steam would be

$$\cdot225 + \cdot02 = \cdot245 \text{ lb. steam.}$$

$$\cdot02 \text{ lb. of sulphur would give } \cdot02 \times 2 = \cdot04 \text{ lb. sulphurous gas.}$$

This quantity is so small that for ordinary purposes it may be neglected.

Then total weights of gases leaving boiler per pound of coal (neglecting aqueous vapour in air and sulphur compounds) are—

Carbonic acid	2·56
Air in excess	7·32
Nitrogen	7·49
Steam	·245
	17·615

As we now know the weight of the different kinds of gases leaving the boiler, and as the temperature of the gases was taken, it is possible to calculate the amount of heat contained in the gases by simply multiplying the weight of

each gas by its specific heat and by its temperature, then add; but perhaps a more convenient way is to first calculate what is known as the "heat capacity," and then simply multiply this by the temperature.

HEAT CAPACITY OF GASES FROM 1 LB. OF COAL.

	Weight.		Specific heat.		Heat capacity.
Carbonic acid	2·56	×	·216	=	0·55
Air....................	7·32	×	·238	=	1·74
Nitrogen	7·49	×	·244	=	1·83
Steam	·24	×	·481	=	·12
					4·24

That is to say, 4·24 units of heat will be absorbed by the flue gases from 1 lb. of coal for each degree rise of temperature. Hence the amount of heat leaving the boiler in the flue gases will be 4·24 multiplied by the temperature. It may here be noted that it is wise to measure all the temperatures from one point, viz., the freezing point of water (32 deg. Fah.).

Let temperature of gases leaving boiler = 732 deg. Fah., then rise of temperature above 32 deg. Fah. = 700 deg. Fah. $\therefore$ heat leaving boiler per lb. of coal

$$= 4{\cdot}24 \times 700 = 2968 \text{ units.}$$

In testing economisers, analyses should be made of the gases leaving, as well as those entering, the economisers, as there is usually considerable difference owing to leakage of air at the economiser chain holes, dampers, brickwork, &c.

Let the air leakage into flues equal 3 lb. of air per pound of coal, then the heat capacity of the gases would be increased by $3 \times {\cdot}238 = {\cdot}714$; therefore total heat capacity of gases leaving the economiser would be $4{\cdot}24 + {\cdot}71 = 4{\cdot}95$.

Let temperature of the gases leaving economiser = 432 deg. Fah., then units of heat per lb. of coal leaving economiser

$$= 4{\cdot}95 \times (432 - 32)$$
$$= 1980.$$

The foregoing calculations would give the number of units of heat leaving both the boiler and the economiser, but when comparing these amounts of heat in a balance sheet with the heat given out by the coal it is necessary to allow for the latent heat of the steam. To do this the

following method might be adopted: Units of heat required to raise moisture in coal (2 per cent) from, say, 60 deg. Fah. to 212 deg. Fah., and evaporate it,

$$= {\cdot}02\,(212 - 60) + {\cdot}02 \times 966$$
$$= 22 \text{ units.}$$

Units required to raise temperature of this steam from 212 deg. to 732 deg. = $(732 - 212)\ {\cdot}02 \times {\cdot}481$ (specific heat of steam) = 5; $\therefore$ total = 22 + 5 = 27.

If moisture be taken as existing in gaseous form, as in previous calculation, the calculation would have been $(732 - 60)\ {\cdot}02 \times {\cdot}481 = 6$.

Therefore, difference (27 − 6 = 21 units) may be considered as heat lost in evaporating moisture in coal.

Ashes and Clinker.

By comparing the actual weight of ash and clinker with the percentage of ash shown by the analysis of the coal, we can determine approximately the amount of unburnt carbon mixed with the ash; and from this the actual quantity of carbon burnt, as used in one of the calculations just dealt with, can be determined.

Heat Supplied to Furnaces.

Seeing that in all our calculations the heat discharged in the flue gases is always reckoned from 32 deg. Fah., it is necessary to allow for the heat contained in the coal and air entering the furnace, in addition to the heat due to the combustion. These allowances can easily be made by remembering that the specific heat of coal is approximately 0·25, and of air 0·238.

A convenient way of summing up the results of tests of boilers and economisers is that of balancing the heat supplied against the heat given to the water and lost in various ways. The form of balance sheet used in ordinary bookkeeping may be adopted for this purpose, as follows: The heat lost by radiation and other causes is obtained by difference, and is made to balance the two sides of the sheets; hence it includes the errors of all the other quantities.

Balance Sheet for Boiler.

Dr.	Heat units.
To calorific value of 1 lb. of dry fuel..............................	....
,, heat contained in fuel thrown into furnaces, reckoning from 32 deg. Fah..	....
,, heat contained in air drawn into furnaces, reckoning from 32 deg. Fah..	....
Total	....

Cr.	Heat units.	Percentage.
By heat transferred to water	....	....
,, heat lost in flue gases, neglecting moisture in coal..	....	....
,, heat lost in evaporating and superheating moisture in coal	....	....
,, heat lost by unburned carbon	....	....
,, heat lost in ashes and clinker	....	....
,, heat lost by radiation and other causes	....	....
Total	....	100

Balance Sheet for Economiser.

Dr.	Heat units.
To heat received in gases from boiler	....
,, heat due to difference of temperature of water in economiser at start and finish of test	....
Total	....

Cr.	Heat units.	Percentage.
By heat transferred to water	....	....
,, heat lost in gases passing to chimney (neglecting that contained in air which leaked in at chain holes and other openings)	....	....
,, heat lost in raising temperature of air which leaked in at chain and other openings	....	....
,, heat lost by radiation and other causes	....	....
Total	....	100

Balance Sheet for Boiler and Economiser.

Dr.	Heat units.
To heat supplied to boilers	....
,, heat due to difference of temperature of water in economiser at start and finish of test	....
Total	....

Cr.	Heat units.	Percentage.
By heat transferred to water..........................	....	....
,, heat in gases passing to chimney (neglecting moisture)	....	....
,, heat used in evaporating and superheating moisture in coal	....	....
,, heat lost by unburned carbon	....	....
,, heat lost n ashes and clinker	....	...
,, heat lost by radiation and other causes	....	....
Total	....	100

ENGINE.

The following list shows the principal results which can be determined from the observations described, and may be useful in showing the points which it is usually desirable to be recorded. Of course, any heading referring to steam jackets will have no application to a non-jacketed engine.

Results of Observations.

1. Average revolutions per minute.
2. Average pressure of steam in supply pipe close to engine.
3. Average pressure of steam in boiler.
4. Average initial pressure in high-pressure cylinder.
5. Average pressure in high-pressure jacket.
6. Average pressure in intermediate jacket.
7. Average pressure in low-pressure jacket.
8. Average vacuum in condenser.
9. Average vacuum in low-pressure cylinder.
10. Average indicated horse power.
11. Average point of cut-off in each cylinder.
12. Average ratio of expansion.
13. Average weight of feed to boilers per indicated horse power per hour.
14. Average net weight of steam supplied to engine per indicated horse power per hour.
15. Average net weight of steam condensed in jackets per indicated horse power per hour.
16. Average weight of coal burnt per indicated horse power per hour.
17. Average weight of steam per indicated horse power, as calculated from diagrams.
18. Average percentage of water present in cylinders at different points of expansion curves, as calculated from diagrams.
19. Average dryness factor of steam supplied to the engine.
20. Average weight of water collected from various drains per indicated horse power per hour.
21. Average quantity of heat supplied to engine per minute.
22. Average quantity of heat discharged to condenser per minute.
23. Average percentage transformed to useful work.
24. Average barometric pressure.

No. 19 would be determined by calculating the weight of steam accounted for by the diagrams, and comparing this with the actual weight of steam and water passed into the cylinders.

The manner in which the percentage of steam fluctuates

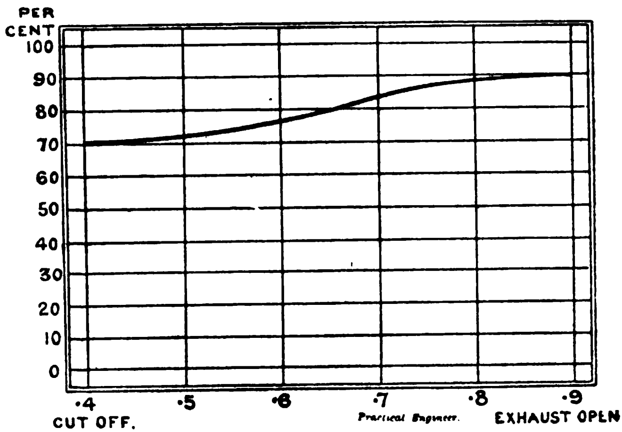


FIG. 1

may be shown very plainly by a diagram similar to that shown at fig. 16.

The remaining results are, on the whole, questions of simple arithmetic, or are the direct results of experiments already explained, and would hardly appear to require further special explanation.

PISTON CONSTANTS.

The following table will be found of great convenience when calculations respecting the indicated horse power of engines are required. The horse power of an engine, as is pointed out on page 52, is obtained by multiplying the speed of the piston in feet per minute by the area of the cylinder in square inches by the average mean pressure on the piston in pounds per square inch, and dividing by 33,000. This may be expressed more briefly as follows :—

$$\frac{\text{Speed} \times \text{area} \times \text{mean pressure}}{33{,}000} = \text{horse power.}$$

In stationary engines it generally happens that the only term in this expression which varies is the mean pressure of the diagram, so that the others may be regarded as a constant quantity. In any case, the area divided by 33,000 is constant, and hence, knowing the speed and mean pressure, the horse power of any engine can be readily determined from the following table almost as quickly as by consulting a table of areas. The table, it will be observed, is arranged on the decimal system, so that the piston constant for any intermediate speed can be easily obtained by simple addition. The value of the table will, perhaps, be best recognised by working an example. For instance, suppose it is required to know the horse power of an engine, the cylinder diameter of which is 23¾ in., and the average mean pressure 30 lb. per square inch, and the piston speed 567 ft. per minute. On consulting the table, it will be seen that

			Ft.		
The piston constant for		......	500	=	6·642
,,	,,	......	60	=	·797
,,	,,	......	7	=	·092
,,	,,	......	567	=	7·531

And the horse power = 7·531 × 30 = 225·93 horse power.

It will be observed that the piston constant for 60 ft. is at once obtained from that for 600 ft. by shifting the decimal point one place, and that for 7 ft by shifting the decimal point for 700 ft. two places.

TABLE OF INDICATED HORSE POWER PER POUND OF MEAN PRESSURE ON PISTON.

Compiled by William H. Fowler, Wh.Sc., Assoc.M.Inst.C.E.

Diameter of cylinder ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
10	0·238	0·476	0·714	0·952	1·190	1·428	1·666	1·904	2·142	2·380
⅛	0·244	0·488	0·732	0·976	1·220	1·464	1·708	1·952	2·196	2·440
¼	0·250	0·500	0·750	1·000	1·250	1·500	1·750	2 000	2·250	2·500
⅜	0·256	0·512	0·769	1·025	1·281	1·537	1·793	2·050	2·306	2·562
½	0·262	0·525	0·787	1·050	1·312	1·574	1·837	2·099	2·362	2·624
⅝	0·269	0·537	0·806	1·075	1·343	1·612	1·881	2·150	2·418	2·687
¾	0·275	0·550	0·825	1·100	1·375	1·650	1·925	2·200	2·475	2·750
⅞	0·281	0·563	0·844	1·126	1·407	1·689	1·970	2·252	2·533	2·815
11	0·288	0·576	0·864	1·152	1·440	1·728	2·016	2·304	2·592	2·880
⅛	0·295	0·589	0·884	1·178	1·473	1·768	2·062	2·357	2·651	2 946
¼	0·301	0·602	0·904	1·205	1·506	1 807	2·108	2·410	2·711	3·012
⅜	0·308	0·616	0 924	1·232	1·539	1·847	2·155	2·463	2·771	3·079
½	0·315	0·630	0·944	1·259	1·574	1·889	2·204	2·518	2·833	3·148
⅝	0·322	0·643	0·965	1·286	1·608	1·930	2·251	2·573	2·894	3·216
¾	0·329	0·657	0·986	1·314	1·643	1·972	2·300	2·629	2·957	3·286
⅞	0·336	0·671	1 007	1·342	1·678	2·014	2·349	2·685	3·020	3·356
12	0·343	0·685	1·028	1·371	1·713	2·056	2·399	2·742	3·084	3·427
⅛	0·350	0·700	1·050	1·400	1·749	2·099	2·449	2·799	3·149	3·499
¼	0·357	0·714	1·071	1·428	1·785	2·143	2·500	2·857	3·214	3·571
⅜	0·364	0·729	1·093	1·458	1·822	2·187	2·551	2·916	3·280	3·645
½	0·372	0·744	1·116	1·488	1·859	2·231	2·603	2·975	3·347	3·719
⅝	0·379	0·759	1·138	1·517	1·896	2·276	2·655	3·034	3·414	3·793
¾	0·387	0·774	1·161	1·548	1·934	2·321	2·708	3·095	3·482	3·869
⅞	0·394	0·789	1·183	1·578	1·972	2·367	2·761	3·156	3·550	3·945

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
13	0·402	0·804	1·207	1·609	2·011	2·413	2·815	3·218	3·620	4·022
⅛	0·410	0·820	1·230	1·640	2·050	2·460	2·870	3·280	3·690	4·100
¼	0·418	0·836	1·253	1·671	2·089	2·507	2·925	3·342	3·760	4·178
⅜	0·426	0·852	1·277	1·703	2·129	2·555	2·981	3·406	3·832	4·258
½	0·434	0·867	1·301	1·735	2·168	2·602	3·036	3·470	3·903	4·337
⅝	0·442	0·884	1·325	1·767	2·209	2·651	3·093	3·534	3·976	4·418
¾	0·450	0·900	1·350	1·800	2·250	2·700	3·150	3·600	4·050	4·500
⅞	0·458	0·916	1·375	1·833	2·291	2·749	3·207	3·666	4·124	4·582
14	0·466	0·933	1·399	1·866	2·332	2·799	3·265	3·732	4·198	4·665
⅛	0·475	0·950	1·424	1·899	2·374	2·849	3·324	3·788	4·273	4·748
¼	0·483	0·967	1·450	1·933	2·416	2·900	3·383	3·866	4·350	4·833
⅜	0·492	0·984	1·475	1·967	2·459	2·951	3·443	3·934	4·426	4·918
½	0·500	1·001	1·501	2·002	2·502	3·002	3·503	4·003	4·504	5·004
⅝	0·509	1·018	1·527	2·036	2·545	3·055	3·564	4·073	4·582	5·091
¾	0·518	1·036	1·553	2·071	2·589	3·107	3·625	4·142	4·660	5·178
⅞	0·527	1·053	1·580	2·106	2·633	3·160	3·686	4·213	4·739	5·266
15	0·535	1·071	1·606	2·142	2·677	3·213	3·748	4·284	4·819	5·355
⅛	0·544	1·089	1·633	2·178	2·722	3·267	3·811	4·356	4·900	5·445
¼	0·553	1·107	1·660	2·214	2·767	3·321	3·874	4·428	4·981	5·535
⅜	0·563	1·125	1·688	2·250	2·813	3·376	3·938	4·501	5·063	5·626
½	0·572	1·144	1·715	2·287	2·859	3·431	4·003	4·574	5·146	5·718
⅝	0·581	1·162	1·743	2·324	2·905	3·486	4·067	4·648	5·229	5·810
¾	0·590	1·181	1·771	2·362	2·952	3·542	4·133	4·723	5·314	5·904
⅞	0·600	1·200	1·799	2·399	2·999	3·599	4·199	4·798	5·398	5·998
16	0·609	1·219	1·828	2·437	3·046	3·656	4·265	4·874	5·484	6·093
⅛	0·619	1·238	1·856	2·475	3·094	3·713	4·332	4·950	5·569	6·188
¼	0·628	1·257	1·885	2·514	3·142	3·771	4·399	5·028	5·656	6·285
⅜	0·638	1·276	1·915	2·553	3·191	3·829	4·467	5·106	5·744	6·382
½	0·648	1·296	1·944	2·592	3·240	3·888	4·536	5·184	5·832	6·480
⅝	0·658	1·316	1·973	2·631	3·289	3·947	4·605	5·262	5·920	6·578
¾	0·668	1·335	2·003	2·671	3·338	4·006	4·674	5·342	6·009	6·677
⅞	0·678	1·355	2·033	2·711	3·388	4·066	4·744	5·422	6·099	6·777

Indicated H.P. per lb of Mean Pressure on Piston—*Continued.*

Diam of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
17	0·688	1·376	2·063	2·751	3·439	4·127	4·815	5·502	6·190	6·878
⅛	0·698	1·396	2·094	2·792	3·490	4·188	4·886	5·584	6·282	6·980
¼	0·708	1·416	2·125	2·833	3·541	4·249	4·957	5·666	6·374	7·082
⅜	0·718	1·437	2·155	2·874	3·592	4·311	5·029	5·748	6·466	7·185
½	0·729	1·458	2·187	2·916	3·644	4·373	5·102	5·831	6·560	7·289
⅝	0·739	1·479	2·218	2·957	3·696	4·436	5·175	5·914	6·654	7·393
¾	0·750	1·500	2·249	2·999	3·749	4·499	5·249	5·998	6·748	7·498
⅞	0·760	1·521	2·281	3·042	3·802	4·562	5·323	6·083	6·844	7·604
18	0·771	1·542	2·313	3·084	3·855	4·627	5·398	6·169	6·940	7·711
⅛	0·782	1·564	2·346	3·128	3·909	4·691	5·473	6·255	7·037	7·819
¼	0·793	1·585	2·378	3·171	3·963	4·756	5·549	6·342	7·134	7·927
⅜	0·804	1·607	2·411	3·214	4·018	4·822	5·625	6·429	7·232	8·036
½	0·815	1·629	2·444	3·258	4·073	4·888	5·702	6·517	7·321	8·146
⅝	0·826	1·651	2·477	3·302	4·128	4·954	5·779	6·605	7·430	8·256
¾	0·837	1·673	2·510	3·347	4·183	5·020	5·857	6·694	7·530	8·367
⅞	0·848	1·696	2·544	3·392	4·239	5·087	5·935	6·783	7·631	8·479
19	0·859	1·718	2·578	3·437	4·296	5·155	6·014	6·874	7·733	8·592
⅛	0·870	1·741	2·611	3·482	4·352	5·223	6·093	6·964	7·834	8·705
¼	0·882	1·764	2·646	3·528	4·409	5·291	6·173	7·055	7·937	8·819
⅜	0·893	1·787	2·680	3·574	4·467	5·360	6·254	7·147	8·041	8·934
½	0·905	1·810	2·715	3·620	4·525	5·430	6·335	7·240	8·145	9·050
⅝	0·917	1·833	2·750	3·666	4·583	5·500	6·416	7·333	8·249	9·166
¾	0·928	1·857	2·785	3·713	4·641	5·570	6·498	7·426	8·355	9·283
⅞	0·940	1·880	2·820	3·760	4·700	5·641	6·581	7·521	8·461	9·401
20	0·952	1·904	2·856	3·808	4·760	5·712	6·664	7·616	8·568	9·520
⅛	0·964	1·928	2·892	3·856	4·819	5·783	6·747	7·711	8·675	9·639
¼	0·976	1·952	2·928	3·904	4·879	5·855	6·831	7·807	8·783	9·759
⅜	0·988	1·976	2·964	3·952	4·940	5·928	6·916	7·904	8·892	9·880
½	1·000	2·000	3·001	4·001	5·001	6·001	7·001	8·002	9·002	10·002
⅝	1·012	2·025	3·037	4·050	5·062	6·074	7·087	8·099	9·112	10·124
¾	1·025	2·049	3·074	4·099	5·123	6·148	7·173	8·198	9·222	10·247
⅞	1·037	2·074	3·111	4·148	5·185	6·223	7·260	8·297	9·334	10·371

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndrins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
21	1·050	2·099	3·149	4·198	5·248	6·298	7·347	8·397	9·446	10·496
⅛	1·062	2·124	3·186	4·248	5·310	6·373	7·435	8·497	9·559	10·621
¼	1·075	2·149	3·224	4·299	5·373	6·448	7·523	8·598	9·672	10·747
⅜	1·087	2·175	3·262	4·350	5·437	6·524	7·612	8·699	9·787	10·874
½	1·100	2·200	3·300	4·400	5·500	6·601	7·701	8·801	9·901	11·001
⅝	1·113	2·226	3·339	4·452	5·565	6·678	7·791	8·904	10·017	11·130
¾	1·126	2·252	3·378	4·504	5·629	6·755	7·881	9·007	10·133	11·259
⅞	1·139	2·278	3·417	4·556	5·694	6·833	7·972	9·111	10·250	11·389
22	1·152	2·304	3·456	4·608	5·759	6·911	8·063	9·215	10·367	11·519
⅛	1·165	2·330	3·495	4·660	5·825	6·990	8·155	9·320	10·485	11·650
¼	1·178	2·356	3·535	4·713	5·891	7·069	8·247	9·426	10·604	11·782
⅜	1·191	2·383	3·574	4·766	5·957	7·149	8·340	9·532	10·723	11·915
½	1·205	2·410	3·615	4·820	6·024	7·229	8·434	9·639	10·844	12·049
⅝	1·218	2·437	3·655	4·878	6·091	7·310	8·528	9·746	10·965	12·183
¾	1·232	2·464	3·695	4·927	6·159	7·391	8·623	9·854	11·086	12·318
⅞	1·245	2·491	3·736	4·982	6·227	7·472	8·718	9·963	11·209	12·454
23	1·259	2·518	3·777	5·036	6·295	7·554	8·813	10·072	11·331	12·590
⅛	1·273	2·545	3·818	5·091	6·363	7·636	8·909	10·182	11·454	12·727
¼	1·286	2·573	3·859	5·146	6·432	7·719	9·005	10·292	11·578	12·865
⅜	1·300	2·601	3·901	5·202	6·502	7·802	9·103	10·403	11·704	13·004
½	1·314	2·629	3·943	5·258	6·572	7·886	9·201	10·515	11·830	13·144
⅝	1·328	2·657	3·985	5·314	6·642	7·970	9·299	10·627	11·956	13·284
¾	1·342	2·685	4·027	5·370	6·712	8·055	9·397	10·740	12·082	13·425
⅞	1·357	2·713	4·070	5·426	6·783	8·140	9·496	10·853	12·209	13·566
24	1·371	2·742	4·113	5·484	6·854	8·225	9·596	10·967	12·338	13·709
⅛	1·385	2·770	4·156	5·541	6·926	8·311	9·696	11·082	12·467	13·852
¼	1·400	2·799	4·199	5·598	6·998	8·398	9·797	11·197	12·596	13·996
⅜	1·414	2·828	4·242	5·656	7·070	8·485	9·899	11·313	12·727	14·141
½	1·429	2·857	4·286	5·714	7·143	8·572	10·000	11·429	12·857	14·286
⅝	1·443	2·886	4·330	5·773	7·216	8·659	10·102	11·546	12·989	14·432
¾	1·458	2·916	4·374	5·832	7·289	8·747	10·205	11·663	13·121	14·579
⅞	1·473	2·945	4·418	5·891	7·363	8·836	10·309	11·782	13·254	14·727

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
25	1·487	2·975	4·462	5·950	7·487	8·925	10·412	11·900	13·387	14·875
⅛	1·502	3·005	4·507	6·010	7·512	9·014	10·517	12·019	13·522	15·024
¼	1·517	3·035	4·552	6·070	7·587	9·104	10·622	12·139	13·657	15·174
⅜	1·532	3·065	4·597	6·130	7·662	9·195	10·727	12·260	13·792	15·325
½	1·548	3·095	4·643	6·190	7·738	9·286	10·833	12·381	13·928	15·476
⅝	1·563	3·126	4·688	6·251	7·814	9·377	10·940	12·502	14·065	15·628
¾	1·578	3·156	4·734	6·312	7·890	9·469	11·047	12·625	14·203	15·781
⅞	1·593	3·187	4·780	6·374	7·967	9·560	11·154	12·747	14·341	15·934
26	1·609	3·218	4·827	6·436	8·044	9·653	11·262	12·871	14·480	16·089
⅛	1·624	3·249	4·873	6·498	8·122	9·746	11·371	12·995	14·620	16·244
¼	1·640	3·280	4·920	6·560	8·200	9·840	11·480	13·120	14·760	16·400
⅜	1·656	3·311	4·967	6·622	8·278	9·934	11·589	13·245	14·900	16·556
½	1·671	3·343	5·014	6·685	8·356	10 028	11·699	13·370	15·042	16·713
⅝	1·687	3·374	5·062	6·749	8·436	10·123	11·810	13·498	15·185	16·872
¾	1·703	3·406	5·109	6·812	8·515	10·218	11·921	13·624	15·327	17·030
⅞	1·719	3·438	5·157	6·876	8·595	10·314	12·033	13·752	15·471	17·190
27	1·735	3·470	5·205	6·940	8·675	10·410	12·145	13·880	15·615	17·250
⅛	1·751	3·502	5·253	7·004	8·755	10·507	12·258	14·009	15·760	17·511
¼	1·767	3·535	5·302	7·069	8·836	10·604	12·371	14·138	15·906	17·673
⅜	1·783	3·567	5·350	7·134	8·917	10·701	12·484	14·268	16·051	17·835
½	1·800	3·600	5·400	7·200	8·999	10·799	12·599	14·399	16·199	17·999
⅝	1·816	3·633	5·449	7·265	9·081	10·898	12·714	14·530	16·347	18·163
¾	1·833	3·665	5·498	7·331	9·163	10·996	12·829	14·662	16·494	18·327
⅞	1·849	3·699	5·548	7·397	9·246	11·096	12·945	14·794	16·644	18·493
28	1·866	3·732	5·598	7·464	9·329	11·195	13·061	14·927	16·793	18·659
⅛	1·883	3·765	5·648	7·530	9·413	11·296	13·178	15·061	16·943	18·826
¼	1·899	3·799	5·698	7·598	9·497	11·396	13·296	15·195	17·095	18·994
⅜	1·916	3·832	5·749	7·665	9·581	11·497	13·413	15·330	17·246	19·162
½	1·933	3·866	5·800	7·733	9·666	11·599	13·532	15·466	17·399	19·332
⅝	1·950	3·900	5·850	7·800	9·750	11·701	13·651	15·601	17·551	19·501
¾	1·967	3·934	5·902	7·869	9·836	11·808	13·770	15·738	17·705	19·672
⅞	1·984	3·969	5·953	7 938	9·922	11·906	13·891	15·875	17·860	19·844

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
29	2·002	4·003	6·005	8·006	10·008	12·010	14·011	16·013	18·014	20·016
⅛	2·019	4·038	6·057	8·076	10·094	12·113	14·132	16·151	18·170	20·189
¼	2·036	4·072	6·109	8·145	10·181	12·217	14·253	16·290	18·326	20·362
⅜	2·054	4·107	6·161	8·215	10·268	12·322	14·376	16·430	18·483	20·537
½	2·071	4·142	6·214	8·285	10·356	12·427	14·498	16·570	18·641	20·712
⅝	2·089	4·178	6·266	8·355	10·444	12·533	14·622	16·710	18·799	20·888
¾	2·106	4·213	6·319	8·426	10·532	12·638	14·745	16·851	18·958	21·064
⅞	2·124	4·248	6·373	8·497	10·621	12·745	14·869	16·994	19·118	21·242
30	2·142	4·284	6·426	8·568	10·710	12·852	14·994	17·136	19·278	21·420
⅛	2·160	4·320	6·480	8·640	10·799	12·959	15·119	17·279	19·439	21·599
¼	2·178	4·356	6·533	8·711	10·889	13·067	15·245	17·422	19·600	21·778
⅜	2·196	4·392	6·588	8·784	10·979	13·175	15·371	17·567	19·763	21·959
½	2·214	4·428	6·642	8·856	11·070	13·284	15·498	17·712	19·926	22·140
⅝	2·232	4·464	6·697	8·929	11·161	13·393	15·625	17·858	20·090	22·322
¾	2·250	4·501	6·751	9·002	11·252	13·502	15·753	18·003	20·254	22·504
⅞	2·269	4·538	6·806	9·075	11·344	13·613	15·882	18·150	20·419	22·688
31	2·287	4·574	6·862	9·149	11·436	13·723	16·010	18·298	20·585	22·872
⅛	2·306	4·611	6·917	9·223	11·528	13·834	16·140	18·446	20·751	23·057
¼	2·324	4·648	6·973	9·297	11·621	13·945	16·269	18·594	20·918	23·242
⅜	2·343	4·686	7·028	9·371	11·714	14·057	16·400	18·742	21·085	23·428
½	2·362	4·723	7·085	9·446	11·808	14·170	16·531	18·893	21·254	23·616
⅝	2·380	4·761	7·141	9·521	11·901	14·282	16·662	19·042	21·423	23·803
¾	2·399	4·798	7·198	9·597	11·996	14·395	16·794	19·194	21·593	23·992
⅞	2·418	4·836	7·254	9·672	12·090	14·509	16·927	19·345	21·763	24·181
32	2·437	4·874	7·311	9·748	12·185	14·623	17·060	19·497	21·934	24·371
⅛	2·456	4·912	7·369	9·825	12·281	14·737	17·193	19·650	22·106	24·562
¼	2·475	4·951	7·426	9·901	12·376	14·852	17·327	19·802	22·278	24·753
⅜	2·495	4·989	7·484	9·978	12·473	14·968	17·462	19·957	22·451	24·946
½	2·514	5·028	7·542	10·056	12·569	15·083	17·597	20·111	22·625	25·139
⅝	2·533	5·066	7·600	10·133	12·666	15·199	17·732	20·266	22·799	25·332
¾	2·553	5·105	7·658	10·211	12·763	15·316	17·869	20·422	22·974	25·527
⅞	2·572	5·144	7·717	10·289	12·861	15·433	18·005	20·578	23·150	25·722

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
33	2·592	5·184	7·775	10·367	12·959	15·551	18·143	20·734	23·326	25·918
⅛	2·611	5·223	7·834	10·446	13·057	15·669	18·280	20·892	23·503	26·115
¼	2·631	5·262	7·894	10·525	13·156	15·787	18·418	21·050	23·681	26·312
⅜	2·651	5·302	7·953	10·604	13·255	15·907	18·558	21·209	23·860	26·511
½	2·671	5·342	8·013	10·684	13·355	16·026	18·697	21·368	24·039	26·710
⅝	2·691	5·382	8·073	10·764	13·454	16·145	18·836	21·527	24·218	26·909
¾	2·711	5·422	8·133	10·844	13·555	16·266	18·977	21·688	24·399	27·110
⅞	2·731	5·462	8·193	10·924	13·655	16·387	19·118	21·849	24·580	27·311
34	2·751	5·503	8·254	11·005	13·756	16·508	19·259	22·010	24·762	27·513
⅛	2·771	5·543	8·314	11·086	13·857	16·629	19·400	22·172	24·943	27·715
¼	2·792	5·584	8·376	11·168	13·959	16·751	19·543	22·335	25·127	27·919
⅜	2·812	5·625	8·437	11·249	14·061	16·874	19·686	22·498	25·311	28·123
½	2·833	5·666	8·498	11·331	14·164	16·997	19·830	22·662	25·495	28·328
⅝	2·853	5·707	8·560	11·414	14·267	17·120	19·974	22·827	25·681	28·534
¾	2·874	5·748	8·622	11·496	14·370	17·244	20·118	22·992	25·866	28·740
⅞	2·895	5·789	8·684	11·579	14·473	17·368	20·268	23·158	26·052	28·947
35	2·915	5·831	8·746	11·662	14·577	17·493	20·408	23·324	26·239	29·155
⅛	2·936	5·873	8·809	11·746	14·682	17·618	20·555	23·491	26·428	29·364
¼	2·957	5·915	8·872	11·829	14·786	17·744	20·701	23·658	26·616	29·573
⅜	2·978	5·957	8·935	11·913	14·891	17·870	20·848	23·826	26·805	29·783
½	2·999	5·999	8·998	11·998	14·997	17·996	20·996	23·995	26·995	29·994
⅝	3·020	6·041	9·061	12·082	15·102	18·123	21·143	24·164	27·184	30·205
¾	3·042	6·084	9·125	12·167	15·209	18·251	21·293	24·334	27·376	30·418
⅞	3·063	6·126	9·189	12·252	15·315	18·379	21·442	24·505	27·568	30·631
36	3·084	6·169	9·253	12·338	15·422	18·507	21·591	24·676	27·760	30·845
⅛	3·106	6·212	9·318	12·424	15·529	18·635	21·741	24·847	27·953	31·059
¼	3·127	6·255	9·382	12·510	15·637	18·765	21·892	25·020	28·147	31·275
⅜	3·149	6·298	9·447	12·596	15·745	18·895	22·044	25·193	28·342	31·491
½	3·171	6·341	9·512	12·683	15·858	19·024	22·195	25·366	28·536	31·707
⅝	3·192	6·385	9·577	12·770	15·962	19·155	22·347	25·540	28·732	31·925
¾	3·214	6·429	9·643	12·857	16·071	19·286	22·500	25·714	28·929	32·143
⅞	3·236	6·472	9·709	12·945	16·181	19·417	22·653	25·890	29·126	32·362

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
37	3·258	6·516	9·775	13·033	16·291	19·549	22·807	26·066	29·324	32·582
⅛	3·280	6·561	9·841	13·121	16·401	19·682	22·962	26·242	29·523	32·803
¼	3·302	6·605	9·907	13·210	16·512	19·814	23·117	26·419	29·722	33·024
⅜	3·325	6·649	9·974	13·298	16·623	19·948	23·272	26·597	29·921	33·246
½	3·347	6·694	10·041	13·388	16·734	20·081	23·428	26·775	30·122	33·469
⅝	3·369	6·738	10·108	13·477	16·846	20·215	23·584	26·954	30·323	33·692
¾	3·392	6·783	10·175	13·566	16·958	20·350	23·741	27·133	30·524	33·916
⅞	3·414	6·828	10·242	13·656	17·070	20·485	23·899	27·313	30·727	34·141
38	3·437	6·873	10·310	13·747	17·183	20·620	24·057	27·494	30·930	34·367
⅛	3·459	6·919	10·378	13·838	17·297	20·756	24·216	27·675	31·135	34·594
¼	3·482	6·964	10·446	13·928	17·410	20·893	24·375	27·857	31·339	34·821
⅜	3·505	7·010	10·515	14·020	17·524	21·029	24·534	28·039	31·544	35·049
½	3·528	7·056	10·583	14·111	17·639	21·167	24·695	28·222	31·750	35·278
⅝	3·551	7·101	10·652	14·203	17·753	21·304	24·855	28·406	31·956	35·507
¾	3·574	7·147	10·721	14·295	17·868	21·442	25·016	28·590	32·163	35·737
⅞	3·597	7·194	10·790	14·387	17·984	21·581	25·178	28·774	32·371	35·968
39	3·620	7·240	10·860	14·480	18·100	21·720	25·340	28·960	32·580	36·200
⅛	3·643	7·286	10·930	14·573	18·216	21·859	25·502	29·146	32·789	36·432
¼	3·666	7·333	10·999	14·666	18·332	21·999	25·665	29·332	32·998	36·665
⅜	3·690	7·380	11·070	14·760	18·449	22·139	25·829	29·519	33·209	36·899
½	3·713	7·427	11·140	14·854	18·567	22·280	25·994	29·707	33·421	37·134
⅝	3·737	7·474	11·211	14·948	18·684	22·421	26·158	29·895	33·632	37·369
¾	3·760	7·521	11·281	15·042	18·802	22·563	26·323	30·084	33·844	37·605
⅞	3·784	7·568	11·353	15·137	18·921	22·705	26·489	30·274	34·058	37·842
40	3·808	7·616	11·424	15·232	19·040	22·848	26·656	30·464	34·272	38·080
⅛	3·832	7·664	11·495	15·327	19·159	22·991	26·823	30·654	34·486	38·318
¼	3·856	7·711	11·567	15·423	19·278	23·134	26·990	30·846	34·701	38·557
⅜	3·880	7·759	11·639	15·519	19·398	23·278	27·158	31·038	34·917	38·797
½	3·904	7·808	11·711	15·615	19·519	23·423	27·327	31·230	35·134	39·038
⅝	3·928	7·856	11·784	15·712	19·639	23·567	27·495	31·428	35·351	39·279
¾	3·952	7·904	11·856	15·808	19·760	23·713	27·665	31·617	35·569	39·521
⅞	3·976	7·953	11·929	15·906	19·882	23·858	27·835	31·811	35·788	39·764

Indicated H.P. per lb of Mean Pressure on Piston—*Continued.*

Diam of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
41	4·001	8·002	12·003	16·003	20·004	24·005	28·006	32·006	36·007	40·008
1/8	4·025	8·050	12·076	16·101	20·126	24·151	28·176	32·202	36·227	40·252
1/4	4·050	8·099	12·149	16·199	20·248	24·298	28·348	32·398	36·447	40·497
3/8	4·074	8·149	12·223	16·297	20·371	24·446	28·520	32·594	36·669	40·743
1/2	4·099	8·198	12·297	16·396	20·495	24·594	28·693	32·792	36·891	40·990
5/8	4·124	8·247	12·371	16·495	20·618	24·742	28·866	32·990	37·113	41·237
3/4	4·148	8·297	12·445	16·594	20·742	24·891	29·039	33·188	37·336	41·485
7/8	4·173	8·347	12·520	16·694	20·867	25·040	29·214	33·387	37·561	41·734
42	4·198	8·397	12·595	16·793	20·991	25·190	29·388	33·586	37·785	41·983
1/8	4·223	8·447	12·670	16·893	21·116	25·340	29·563	33·786	38·010	42·233
1/4	4·248	8·497	12·745	16·994	21·242	25·490	29·739	33·987	38·236	42·484
3/8	4·274	8·547	12·821	17·094	21·368	25·642	29·915	34·189	38·462	42·736
1/2	4·299	8·598	12·897	17·196	21·494	25·793	30·092	34·391	38·690	42·989
5/8	4·324	8·648	12·973	17·297	21·621	25·945	30·269	34·594	38·918	43·242
3/4	4·350	8·699	13·049	17·398	21·748	26·098	30·447	34·797	39·146	43·496
7/8	4·375	8·750	13·125	17·500	21·875	26·251	30·626	35·001	39·376	43·751
43	4·401	8·801	13·202	17·602	22·003	26·404	30·804	35·205	39·605	44·006
1/8	4·426	8·852	13·279	17·705	22·131	26·557	30·983	35·410	39·836	44·262
1/4	4·452	8·904	13·356	17·808	22·259	26·711	31·163	35·615	40·067	44·519
3/8	4·478	8·955	13·433	17·911	22·388	26·866	31·344	35·822	40·299	44·777
1/2	4·504	9·007	13·511	18·014	22·518	27·022	31·525	36·029	40·532	45·036
5/8	4·529	9·059	13·588	18·118	22·647	27·177	31·706	36·236	40·765	45·295
3/4	4·555	9·111	13·666	18·222	22·777	27·333	31·888	36·444	40·999	45·555
7/8	4·581	9·163	13·744	18·326	22·907	27·489	32·070	36·652	41·233	45·815
44	4·608	9·215	13·823	18·431	23·038	27·646	32·254	36·862	41·469	46·077
1/8	4·634	9·268	13·902	18·536	23·169	27·803	32·437	37·071	41·705	46·339
1/4	4·660	9·320	13·981	18·641	23·301	27·961	32·621	37·282	41·942	46·602
3/8	4·686	9·373	14·059	18·746	23·432	28·119	32·805	37·492	42·178	46·865
1/2	4·713	9·426	14·139	18·852	23·565	28·278	32·991	37·704	42·417	47·130
5/8	4·739	9·479	14·218	18·958	23·697	28·437	33·176	37·916	42·655	47·395
3/4	4·766	9·532	14·298	19·064	23·830	28·597	33·363	38·129	42·895	47·661
7/8	4·793	9·586	14·378	19·171	23·964	28·757	33·550	38·342	43·135	47·928

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
45	4·819	9·639	14·458	19·278	24·097	28·917	33·736	38·556	43·375	48·195
⅛	4·846	9·693	14·539	19·385	24·231	29·078	33·924	38·770	43·617	48·463
¼	4·873	9·746	14·620	19·493	24·366	29·239	34·112	38·986	43·859	48·732
⅜	4·900	9·800	14·700	19·600	24·500	29·401	34·301	39·201	44·101	49·001
½	4·927	9·854	14·782	19·709	24·636	29·563	34·490	39·418	44·345	49·272
⅝	4·954	9·909	14·863	19·817	24·771	29·726	34·680	39·634	44·589	49·543
¾	4·981	9·963	14·944	19·926	24·907	29·889	34·870	39·852	44·833	49·815
⅞	5·009	10·017	15·026	20·035	25·043	30·052	35·061	40·070	45·078	50·087
46	5·036	10·072	15·108	20·144	25·180	30·217	35·253	40·289	45·325	50·361
⅛	5·063	10·127	15·190	20·254	25·317	30·381	35·444	40·508	45·571	50·635
¼	5·091	10·182	15·273	20·364	25·455	30·546	35·637	40·728	45·819	50·910
⅜	5·118	10·237	15·355	20·474	25·592	30·711	35·829	40·948	46·066	51·185
½	5·146	10·292	15·439	20·585	25·731	30·877	36·023	41·170	46·316	51·462
⅝	5·174	10·348	15·522	20·696	25·869	31·043	36·217	41·391	46·565	51·739
¾	5·202	10·403	15·605	20·806	26·008	31·210	36·411	41·613	46·814	52·016
⅞	5·229	10·459	15·688	20·918	26·147	31·377	36·606	41·836	47·065	52·295
47	5·257	10·515	15·772	21·030	26·287	31·544	36·802	42·059	47·317	52·574
⅛	5·285	10·571	15·856	21·142	26·427	31·712	36·998	42·283	47·569	52·854
¼	5·313	10·627	15·940	21·254	26·567	31·881	37·194	42·508	47·821	53·135
⅜	5·342	10·683	16·025	21·366	26·708	32·050	37·391	42·733	48·074	53·416
½	5·370	10·740	16·110	21·480	26·849	32·219	37·589	42·959	48·329	53·699
⅝	5·398	10·796	16·195	21·593	26·991	32·389	37·787	43·186	48·584	53·982
¾	5·426	10·853	16·279	21·706	27·132	32·559	37·985	43·412	48·838	54·265
⅞	5·455	10·910	16·365	21·820	27·275	32·730	38·185	43·640	49·095	54·55[illegible]
48	5·483	10·967	16·450	21·934	27·417	32·901	38·384	43·868	49·351	54·835
⅛	5·512	11·024	16·536	22·048	27·560	33·073	38·585	44·097	49·609	55·121
¼	5·541	11·082	16·622	22·163	27·704	33·245	38·786	44·326	49·867	55·408
⅜	5·569	11·139	16·708	22·278	27·847	33·417	38·986	44·556	50·125	55·695
½	5·598	11·197	16·795	22·394	27·992	33·590	39·189	44·787	50·386	55·984
⅝	5·627	11·254	16·882	22·509	28·136	33·763	39·390	45·018	50·645	56·272
¾	5·656	11·312	16·969	22·625	28·281	33·937	39·593	45·250	50·906	56·562
⅞	5·685	11·371	17·056	22·741	28·426	34·112	39·797	45·482	51·168	56·853

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet Per Minute.									
	100	200	300	400	500	600	700	800	900	1000
49	5·714	11·429	17·143	22·858	28·572	34·286	40·001	45·715	51·430	57·144
⅛	5·744	11·487	17·231	22·974	28·718	34·462	40·205	45·949	51·692	57·436
¼	5·773	11·546	17·318	23·091	28·864	34·637	40·410	46·182	51·955	57·728
⅜	5·802	11·604	17·406	23·208	29·010	34·813	40·615	46·417	52·219	58·021
½	5·832	11·668	17·495	23·326	29·158	34·990	40·821	46·653	52·484	58·316
⅝	5·861	11·722	17·583	23·444	29·305	35·167	41·028	46·889	52·750	58·611
¾	5·891	11·781	17·672	23·562	29·453	35·344	41·234	47·125	53·015	58·906
⅞	5·920	11·841	17·761	23·681	29·601	35·522	41·442	47·362	53·283	59·203
50	5·950	11·900	17·850	23·800	29·750	35·700	41·650	47·600	53·550	59·500
⅛	5·980	11·960	17·940	23·920	29·900	35·880	41·860	47·840	53·820	59·800
¼	6·010	12·019	18·029	24·038	30·048	36·058	42·067	48·077	54·086	60·096
⅜	6·040	12·079	18·119	24·158	30·198	36·238	42·277	48·317	54·356	60·396
½	6·070	12·139	18·209	24·278	30·348	36·418	42·487	48·557	54·626	60·696
⅝	6·100	12·199	18·299	24·399	30·498	36·598	42·698	48·798	54·897	60·997
¾	6·130	12·260	18·389	24·519	30·649	36·779	42·909	49·038	55·168	61·298
⅞	6·160	12·320	18·480	24·640	30·800	36·961	43·121	49·281	55·441	61·601
51	6·190	12·381	18·571	24·762	30·952	37·142	43·333	49·523	55·714	61·904
⅛	6·221	12·442	18·662	24·883	31·104	37·325	43·546	49·766	55·987	62·208
¼	6·251	12·502	18·754	25·005	31·256	37·507	43·758	50·010	56·261	62·512
⅜	6·282	12·563	18·845	25·127	31·408	37·690	43·972	50·254	56·535	62·817
½	6·312	12·625	18·937	25·250	31·562	37·874	44·187	50·499	56·812	63·124
⅝	6·343	12·686	19·029	25·372	31·715	38·058	44·410	50·744	57·087	63·430
¾	6·374	12·748	19·121	25·495	31·869	38·243	44·617	50·990	57·364	63·738
⅞	6·405	12·809	19·214	25·618	32·023	38·428	44·832	51·237	57·641	64·046
52	6·435	12·871	19·306	25·742	32·177	38·613	45·048	51·484	57·919	64·355
⅛	6·466	12·933	19·399	25·866	32·332	38·799	45·265	51·732	58·198	64·665
¼	6·497	12·995	19·492	25·990	32·487	38·985	45·482	51·980	58·477	64·975
⅜	6·529	13·057	19·586	26·115	32·643	39·172	45·701	52·230	58·758	65·287
½	6·560	13·120	19·680	26·240	32·799	39·359	45·919	52·479	59·039	65·599
⅝	6·591	13·182	19·773	26·364	32·955	39·547	46·138	52·729	59·310	65·911
¾	6·622	13·245	19·867	26·490	33·112	39·735	46·357	52·970	59·602	66·225
⅞	6·654	13·308	19·962	26·616	33·269	39·923	46·577	53·231	59·885	66·5[illegible]9

Indicated H.P. per lb. of Mean Pressure on Piston – *Continued*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
53	6·685	13·371	20·056	26·742	33·427	40·112	46·798	53·483	60·169	66·854
⅛	6·717	13·434	20·151	26·868	33·585	40·202	47·019	53·736	60·453	67·170
¼	6·749	13·497	20·246	26·994	33·743	40·492	47·240	53·989	60·737	67·486
⅜	6·780	13·561	20·341	27·122	33·902	40·682	47·463	54·243	61·024	67·804
½	6·812	13·624	20·437	27·249	34·061	40·873	47·685	54·498	61·310	68·122
⅝	6·844	13·688	20·532	27·376	34·220	41·064	47·908	54·752	61·596	68·440
¾	6·876	13·752	20·628	27·504	34·380	41·256	48·132	55·008	61·884	68·760
⅞	6·908	13·816	20·724	27·632	34·540	41·448	48·356	55·264	62·172	69·080
54	6·940	13·880	20·820	27·760	34·700	41·641	48·581	55·521	62·461	69·401
⅛	6·972	13·944	20·917	27·889	34·861	41·833	48·805	55·778	62·750	69·772
¼	7·004	14·009	21·013	28·018	35·022	42·027	49·081	56·036	63·040	70·045
⅜	7·037	14·074	21·110	28·147	35·184	42·221	49·258	56·294	63·331	70·368
½	7·069	14·138	21·208	28·277	35·346	42·415	49·484	56·554	63·623	70·692
⅝	7·102	14·203	21·305	28·407	35·508	42·610	49·712	56·814	63·915	71·017
¾	7·134	14·268	21·403	28·537	35·671	42·805	49·939	57·074	64·208	71·342
⅞	7·167	14·334	21·500	28·667	35·834	43·001	50·168	57·334	64·501	71·668
55	7·199	14·399	21·598	28·798	35·997	43·197	50·396	57·596	64·795	71·995
⅛	7·232	14·465	21·697	28·929	36·161	43·394	50·626	57·858	65·091	72·323
¼	7·265	14·530	21·795	29·060	36·325	43·591	50·856	58·121	65·386	72·651
⅜	7·298	14·596	21·894	29·192	36·490	43·788	51·086	58·384	65·682	72·980
½	7·331	14·662	21·993	29·324	36·655	43·986	51·317	58·648	65·979	73·310
⅝	7·364	14·728	22·093	29·457	36·821	44·185	51·549	58·914	66·278	73·642
¾	7·397	14·704	22·192	29·589	36·986	44·383	51·780	59·178	66·575	73·972
⅞	7·430	14·861	22·291	29·722	37·152	44·582	52·013	59·443	66·874	74·304
56	7·464	14·927	22·391	29·855	37·318	44·782	52·246	59·710	67·173	74·637
⅛	7·497	14·994	22·491	29·988	37·485	44·982	52·479	59·976	67·473	74·970
¼	7·530	15·061	22·591	30·122	37·652	45·183	52·713	60·244	67·774	75·305
⅜	7·564	15·128	22·692	30·256	37·820	45·384	52·948	60·512	68·076	75·640
½	7·598	15·195	22·793	30·390	37·988	45·586	53·183	60·781	68·378	75·976
⅝	7·631	15·262	22·894	30·525	38·156	45·787	53·418	61·050	68·681	76·312
¾	7·665	15·320	22·995	30·660	38·324	45·989	53·654	61·319	68·984	76·649
⅞	7·699	15·397	23·096	30·795	38·493	46·192	53·891	61·590	69·288	76·987

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
57	7·783	15·465	23·198	30·930	38·663	46·396	54·128	61·861	69·593	77·326
⅛	7·767	15·533	23·300	31·066	38·833	46·600	54·366	62·133	69·899	77·666
¼	7·801	15·601	23·402	31·202	39·003	46·804	54·604	62·405	70·205	78·006
⅜	7·835	15·669	23·504	31·339	39·173	47·008	54·843	62·678	70·512	78·347
½	7·869	15·738	23·607	31·476	39·344	47·213	55·082	62·951	70·820	78·689
⅝	7·903	15·806	23·709	31·612	39·515	47·419	55·322	63·225	71·128	79·031
¾	7·937	15·875	23·812	31·750	39·687	47·624	55·562	63·499	71·437	79·374
⅞	7·972	15·944	23·915	31·887	39·859	47·831	55·803	63·774	71·746	79·718
58	8·006	16·013	24·019	32·025	40·031	48·038	56·044	64·050	72·057	80·063
⅛	8·041	16·082	24·123	32·164	40·204	48·245	56·286	64·327	72·368	80·409
¼	8·075	16·151	24·226	32·302	40·377	48·453	56·528	64·604	72·679	80·755
⅜	8·110	16·220	24·331	32·441	40·551	48·661	56·771	64·882	72·992	81·102
½	8·145	16·290	24·435	32·580	40·725	48·870	57·015	65·160	73·305	81·450
⅝	8·180	16·360	24·539	32·719	40·899	49·079	57·259	65·438	73·618	81·798
¾	8·215	16·429	24·644	32·859	41·073	49·288	57·503	65·718	73·932	82·147
⅞	8·250	16·499	24·749	32·999	41·248	49·498	57·748	65·998	74·247	82·497
59	8·285	16·570	24·854	33·139	41·424	49·709	57·994	66·278	74·563	82·848
⅛	8·320	16·640	24·960	33·280	41·599	49·919	58·239	66·559	74·879	83·199
¼	8·355	16·710	25·065	33·420	41·775	50·131	58·486	66·841	75·196	83·551
⅜	8·390	16·781	25·171	33·562	41·952	50·342	58·733	67·123	75·514	83·904
½	8·426	16·852	25·277	33·703	42·129	50·555	58·981	67·406	75·832	84·258
⅝	8·461	16·922	25·384	33·845	42·306	50·767	59·228	67·690	76·151	84·612
¾	8·497	16·993	25·490	33·987	42·483	50·980	59·477	67·974	76·470	84·967
⅞	8·532	17·065	25·597	34·129	42·661	51·194	59·726	68·258	76·791	85·323
60	8·568	17·136	25·704	34·272	42·840	51·408	59·976	68·544	77·112	85·680
⅛	8·604	17·207	25·811	34·415	43·018	51·622	60·226	68·830	77·433	86·037
¼	8·639	17·279	25·918	34·558	43·197	51·837	60·476	69·116	77·755	86·395
⅜	8·675	17·351	26·026	34·702	43·377	52·052	60·728	69·403	78·079	86·754
½	8·711	17·423	26·134	34·846	43·557	52·268	60·980	69·691	78·403	87·114
⅝	8·747	17·495	26·242	34·990	43·737	52·484	61·232	69·979	78·727	87·474
¾	8·783	17·567	26·350	35·134	43·917	52·701	61·484	70·268	79·051	87·835
⅞	8·820	17·639	26·459	35·279	44·098	52·918	61·738	70·558	79·377	88·197

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
61	8·856	17·712	26·568	35·424	44·280	53·136	61·992	70·848	79·704	88·560
⅛	8·892	17·785	26·677	35·569	44·461	53·354	62·246	71·138	80·031	88·923
¼	8·929	17·857	26·786	35·715	44·643	53·572	62·501	71·430	80·358	89·287
⅜	8·965	17·930	26·896	35·861	44·826	53·791	62·756	71·722	80·687	89·652
½	9·002	18·004	27·005	36·007	45·009	54·011	63·013	72·014	81·016	90·018
⅝	9·038	18·077	27·115	36·154	45·192	54·230	63·269	72·307	81·346	90·384
¾	9·075	18·150	27·225	36·300	45·375	54·451	63·526	72·601	81·676	90·751
⅞	9·112	18·224	27·336	36·448	45·559	54·671	63·783	72·895	82·007	91·119
62	9·149	18·297	27·446	36·595	45·743	54·892	64·041	73·190	82·338	91·487
⅛	9·186	18·371	27·557	36·742	45·928	55·114	64·299	73·485	82·670	91·856
¼	9·223	18·445	27·668	36·890	46·113	55·336	64·558	73·781	83·003	92·226
⅜	9·260	18·519	27·779	37·039	46·298	55·558	64·818	74·078	83·337	92·597
½	9·297	18·594	27·891	37·188	46·484	55·781	65·078	74·375	83·572	92·969
⅝	9·334	18·668	28·002	37·336	46·670	56·005	65·339	74·673	84·007	93·341
¾	9·371	18·743	28·114	37·486	46·857	56·228	65·600	74·971	84·343	93·714
⅞	9·409	18·818	28·226	37·635	47·044	56·453	65·862	75·270	84·679	94·088
63	9·446	18·892	28·339	37·785	47·231	56·677	66·123	75·570	85·016	94·462
⅛	9·484	18·967	28·451	37·935	47·418	56·902	66·386	75·870	85·353	94·837
¼	9·521	19·043	28·564	38·085	47·606	57·128	66·649	76·170	85·692	95·213
⅜	9·559	19·118	28·677	38·236	47·795	57·354	66·913	76·472	86·031	95·590
½	9·597	19·194	28·790	38·387	47·984	57·581	67·178	76·774	86·371	95·968
⅝	9·635	19·269	28·904	38·538	48·173	57·808	67·442	77·077	86·771	96·346
¾	9·672	19·345	29·017	38·690	48·362	58·035	67·707	77·380	87·052	96·725
⅞	9·710	19·421	29·131	38·842	48·552	58·262	67·973	77·683	87·394	97·104
64	9·748	19·497	29·245	38·994	48·742	58·491	68·239	77·988	87·736	97·485
⅛	9·787	19·573	29·360	39·146	48·933	58·720	68·506	78·293	88·079	97·866
¼	9·825	19·650	29·474	39·299	49·124	58·949	68·774	78·598	88·423	98·248
⅜	9·863	19·726	29·589	39·452	49·315	59·179	69·042	78·905	88·768	98·631
½	9·901	19·803	29·704	39·606	49·507	59·408	69·310	79·211	89·113	99·014
⅝	9·940	19·880	29·819	39·759	49·699	59·639	69·579	79·518	89·458	99·398
¾	9·978	19·957	29·935	39·913	49·891	59·870	69·848	79·826	89·805	99·783
⅞	10·017	20·034	30·051	40·068	50·085	60·102	70·119	80·136	90·153	100·17

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
65	10·065	20·110	30·165	40·220	50·275	60·330	70·385	80·440	90·495	100·55
⅛	10·094	20·188	30·282	40·376	50·470	60·564	70·658	80·752	90·846	100·94
¼	10·133	20·266	30·399	40·532	50·665	60·798	70·931	81·064	91·197	101·33
⅜	10·172	20·344	30·516	40·688	50·860	61·032	71·204	81·376	91·548	101·72
½	10·211	20·422	30·633	40·844	51·055	61·266	71·477	81·688	91·899	102·11
⅝	10·250	20·500	30·750	41·000	51·250	61·500	71·750	82·000	92·250	102·50
¾	10·289	20·578	30·867	41·156	51·445	61·734	72·023	82·312	92·601	102·89
⅞	10·328	20·656	30·984	41·312	51·640	61·968	72·296	82·624	92·952	103·28
66	10·367	20·734	31·101	41·468	51·835	62·202	72·569	82·936	93·303	103·67
⅛	10·407	20·814	31·221	41·628	52·035	62·442	72·849	83·256	93·663	104·07
¼	10·446	20·892	31·338	41·784	52·230	62·676	73·122	83·568	94·014	104·46
⅜	10·485	20·970	31·455	41·940	52·425	62·910	73·395	83·880	94·365	104·85
½	10·525	21·050	31·575	42·100	52·625	63·150	73·675	84·200	94·725	105·25
⅝	10·565	21·130	31·695	42·260	52·825	63·390	73·955	84·520	95·085	105·65
¾	10·604	21·208	31·812	42·416	53·020	63·624	74·228	84·832	95·436	106·04
⅞	10·644	21·288	31·932	42·576	53·220	63·864	74·508	85·152	95·796	106·44
67	10·684	21·368	32·052	42·736	53·420	64·104	74·788	85·472	96·156	106·84
⅛	10·724	21·448	32·172	42·896	53·620	64·344	75·068	85·792	96·516	107·24
¼	10·764	21·528	32·292	43·056	53·820	64·584	75·348	86·112	96·876	107·64
⅜	10·804	21·608	32·412	43·216	54·020	64·824	75·628	86·432	97·326	108·04
½	10·844	21·688	32·532	43·376	54·220	65·064	75·908	86·752	97·596	108·44
⅝	10·884	21·768	32·652	43·536	54·420	65·304	76·188	87·072	97·956	108·84
¾	10·924	21·848	32·772	43·696	54·620	65·544	76·468	87·392	98·316	109·24
⅞	10·965	21·930	32·895	43·860	54·825	65·790	76·755	87·720	98·685	109·65
68	11·005	22·010	33·015	44·020	55·025	66·030	77·035	88·040	99·045	110·05
⅛	11·046	22·092	33·138	44·184	55·230	66·276	77·322	88·368	99·414	110·46
¼	11·086	22·172	33·258	44·344	55·430	66·516	77·602	88·688	99·774	110·86
⅜	11·127	22·254	33·381	44·508	55·635	66·762	77·889	89·016	100·14	111·27
½	11·168	22·336	33·504	44·672	55·840	67·008	78·176	89·344	100·51	111·68
⅝	11·208	22·416	33·624	44·832	56·040	67·248	78·456	89·664	100·87	112·08
¾	11·249	22·498	33·747	44·996	56·245	67·494	78·743	89·992	101·24	112·49
⅞	11·290	22·580	33·870	45·160	56·450	67·740	79·030	90·320	101·61	112·90

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
69	11·331	22·662	33·993	45·324	56·655	67·986	79·317	90·648	101·98	113·31
⅛	11·372	22·744	34·116	45·488	56·860	68·232	79·604	90·976	102·35	113·72
¼	11·413	22·826	34·239	45·652	57·065	68·478	79·891	91·304	102·72	114·13
⅜	11·455	22·910	34·365	45·820	57·275	68·730	80·185	91·640	103·09	114·55
½	11·496	22·992	34·488	45·984	57·480	68·976	80·472	91·968	103·46	114·96
⅝	11·537	23·074	34·611	46·148	57·685	69·222	80·759	92·296	103·83	115·37
¾	11·579	23·158	34·737	46·316	57·895	69·474	81·053	92·632	104·21	115·79
⅞	11·620	23·240	34·860	46·480	58·100	69·720	81·340	92·960	104·58	116·20
70	11·662	23·324	34·986	46·648	58·310	69·972	81·634	93·296	104·96	116·62
⅛	11·704	23·408	35·112	46·816	58·520	70·224	81·928	93·632	105·34	117·04
¼	11·745	23·490	35·235	46·980	58·725	70·470	82·215	93·960	105·70	117·45
⅜	11·787	23·574	35·361	47·148	58·935	70·722	82·509	94·296	106·08	117·87
½	11·829	23·658	35·487	47·316	59·145	70·974	82·803	94·632	106·46	118·29
⅝	11·871	23·742	35·613	47·484	59·355	71·226	83·097	94·968	106·84	118·71
¾	11·913	23·826	35·739	47·652	59·565	71·478	83·391	95·304	107·22	119·13
⅞	11·955	23·910	35·865	47·820	59·775	71·730	83·685	95·640	107·59	119·55
71	11·998	23·996	35·994	47·992	59·990	71·988	83·986	95·984	107·98	119·98
⅛	12·040	24·080	36·120	48·160	60·200	72·240	84·280	96·320	108·36	120·40
¼	12·082	24·164	36·246	48·328	60·410	72·492	84·574	96·656	108·74	120·82
⅜	12·125	24·250	36·375	48·500	60·625	72·750	84·875	97·000	109·12	121·25
½	12·167	24·334	36·501	48·668	60·835	73·002	85·169	97·336	109·50	121·67
⅝	12·210	24·420	36·630	48·840	61·050	73·260	85·470	97·680	109·89	122·10
¾	12·252	24·504	36·756	49·008	61·260	73·512	85·764	98·016	110·27	122·52
⅞	12·295	24·590	36·885	49·180	61·475	73·770	86·065	98·360	110·65	122·95
72	12·338	24·676	37·014	49·352	61·690	74·028	86·366	98·704	111·04	123·38
⅛	12·381	24·762	37·143	49·524	61·905	74·286	86·667	99·048	111·43	123·81
¼	12·424	24·848	37·272	49·696	62·120	74·544	86·968	99·392	111·82	124·24
⅜	12·467	24·934	37·401	49·868	62·335	74·802	87·269	99·736	112·20	124·67
½	12·510	25·020	37·530	50·040	62·550	75·060	87·570	100·08	112·59	125·10
⅝	12·553	25·106	37·659	50·212	62·765	75·318	87·871	100·42	112·98	125·53
¾	12·596	25·192	37·788	50·384	62·980	75·576	88·172	100·77	113·36	125·96
⅞	12·640	25·280	37·920	50·560	63·200	75·840	88·480	101·12	113·76	126·40

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
73	12·683	25·366	38·049	50·732	63·415	76·098	88·781	101·46	114·15	126·83
⅛	12·726	25·452	38·178	50·904	63·630	76·356	89·082	101·81	114·53	127·26
¼	12·770	25·540	38·310	51·080	63·850	76·620	89·390	102·16	114·93	127·70
⅜	12·814	25·628	38·442	51·256	64·070	76·884	89·698	102·51	115·33	128·14
½	12·857	25·714	38·571	51·428	64·285	77·142	89·997	102·86	115·71	128·57
⅝	12·901	25·802	38·703	51·604	64·505	77·406	90·307	103·21	116·11	129·01
¾	12·945	25·890	38·835	51·780	64·725	77·670	90·615	103·56	116·50	129·45
⅞	12·989	25·978	38·967	51·956	64·945	77·934	90·923	103·91	116·90	129·89
74	13·033	26·066	39·099	52·132	65·165	78·198	91·231	104·26	117·30	130·33
⅛	13·077	26·154	39·231	52·308	65·385	78·462	91·539	104·62	117·69	130·77
¼	13·121	26·242	39·363	52·484	65·605	78·726	91·847	104·97	118·09	131·21
⅜	13·165	26·330	39·495	52·660	65·825	78·990	92·155	105·32	118·48	131·65
½	13·210	26·420	39·630	52·840	66·050	79·260	92·470	105·68	118·89	132·10
⅝	13·254	26·508	39·762	53·016	66·270	79·524	92·778	106·03	119·29	132·54
¾	13·298	26·596	39·894	53·192	66·490	79·788	93·086	106·38	119·68	132·98
⅞	13·343	26·686	40·029	53·372	66·715	80·058	93·401	106·74	120·09	133·43
75	13·387	26·774	40·161	53·548	66·935	80·322	93·709	107·10	120·48	133·87
⅛	13·432	26·864	40·296	53·728	67·160	80·592	94·024	107·46	120·89	134·32
¼	13·477	26·954	40·431	53·908	67·385	80·862	94·339	107·82	121·29	134·77
⅜	13·522	27·044	40·566	54·088	67·610	81·132	94·654	108·18	121·70	135·22
½	13·567	27·134	40·701	54·268	67·835	81·402	94·969	108·54	122·10	135·67
⅝	13·612	27·224	40·836	54·448	68·060	81·672	95·284	108·90	122·51	136·12
¾	13·657	27·314	40·971	54·628	68·285	81·942	95·599	109·26	122·91	136·57
⅞	13·702	27·404	41·106	54·808	68·510	82·212	95·914	109·62	123·32	137·02
76	13·747	27·494	41·241	54·988	68·735	82·482	96·229	109·98	123·72	137·47
⅛	13·792	27·584	41·376	55·168	68·960	82·752	96·544	110·34	124·13	137·92
¼	13·837	27·674	41·511	55·348	69·185	83·022	96·859	110·70	124·53	138·37
⅜	13·883	27·766	41·649	55·532	69·415	83·298	97·181	111·06	124·95	138·83
½	13·928	27·856	41·784	55·712	69·640	83·568	97·496	111·42	125·35	139·28
⅝	13·974	27·948	41·922	55·896	69·870	83·844	97·818	111·79	125·77	139·74
¾	14·020	28·040	42·060	56·080	70·100	84·120	98·140	112·16	126·18	140·20
⅞	14·065	28·130	42·195	56·260	70·325	84·390	98·455	112·52	126·58	140·65

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet Per Minute.									
	100	200	300	400	500	600	700	800	900	1000
77	14·111	28·222	42·333	56·444	70·555	84·666	98·777	112·89	127·00	141·11
⅛	14·157	28·314	42·471	56·628	70·785	84·942	99·099	113·26	127·41	141·57
¼	14·203	28·406	42·609	56·812	71·015	85·218	99·421	113·62	127·83	142·03
⅜	14·249	28·498	42·747	56·996	71·245	85·494	99·743	113·99	128·24	142·49
½	14·295	28·590	42·885	57·180	71·475	85·770	100·06	114·36	128·65	142·95
⅝	14·341	28·682	43·023	57·364	71·705	86·046	100·39	114·73	129·07	143·41
¾	14·387	28·774	43·161	57·548	71·935	86·322	100·71	115·10	129·48	143·87
⅞	14·434	28·868	43·302	57·736	72·170	86·604	101·04	115·47	129·91	144·34
78	14·480	28·960	43·440	57·920	72·400	86·880	101·36	115·84	130·32	144·80
⅛	14·526	29·052	43·578	58·104	72·630	87·156	101·68	116·21	130·73	145·26
¼	14·573	29·146	43·719	58·292	72·865	87·438	102·01	116·58	131·16	145·73
⅜	14·619	29·238	43·857	58·476	73·095	87·714	102·33	116·95	131·57	146·19
½	14·666	29·332	43·998	58·664	73·330	87·996	102·66	117·33	131·99	146·66
⅝	14·713	29·426	44·139	58·852	73·565	88·278	102·99	117·70	132·42	147·13
¾	14·760	29·520	44·280	59·040	73·800	88·560	103·32	118·08	132·84	147·60
⅞	14·807	29·614	44·421	59·228	74·035	88·842	103·65	118·46	133·26	148·07
79	14·854	29·708	44·562	59·416	74·270	89·124	103·98	118·83	133·69	148·54
⅛	14·901	29·802	44·703	59·604	74·505	89·406	104·31	119·21	134·11	149·01
¼	14·948	29·896	44·844	59·792	74·740	89·688	104·64	119·58	134·53	149·48
⅜	14·995	29·990	44·985	59·980	74·975	89·970	104·96	119·96	134·95	149·95
½	15·042	30·084	45·126	60·168	75·210	90·252	105·29	120·34	135·38	150·42
⅝	15·090	30·180	45·270	60·360	75·450	90·540	105·63	120·72	135·81	150·90
¾	15·137	30·274	45·411	60·548	75·685	90·822	105·96	121·10	136·23	151·37
⅞	15·184	30·368	45·552	60·736	75·920	91·104	106·29	121·47	136·66	151·84
80	15·232	30·464	45·696	60·928	76·160	91·392	106·62	121·86	137·09	152·32
⅛	15·280	30·560	45·840	61·120	76·400	91·680	106·96	122·24	137·52	152·36
¼	15·327	30·654	45·981	61·308	76·635	91·962	107·29	122·62	137·94	153·27
⅜	15·375	30·750	46·125	61·500	76·875	92·250	107·62	123·00	138·37	153·75
½	15·423	30·846	46·269	61·692	77·115	92·538	107·96	123·38	138·81	154·23
⅝	15·471	30·942	46·413	61·884	77·355	92·826	108·30	123·77	139·24	154·71
¾	15·519	31·038	46·557	62·076	77·595	93·114	108·63	124·15	139·67	155·19
⅞	15·567	31·134	46·701	62·268	77·835	93·402	108·97	124·54	140·10	155·67

Indicated H.P. per lb of Mean Pressure on Piston—*Continued.*

Diam of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
81	15·615	31·230	46·845	62·460	78·075	93·690	109·30	124·92	140·53	156·15
⅛	15·663	31·326	46·989	62·652	78·315	93·978	109·64	125·30	140·97	156·63
¼	15·712	31·424	47·136	62·848	78·560	94·272	109·98	125·70	141·41	157·12
⅜	15·760	31·520	47·280	63·040	78·800	94·560	110·32	126·08	141·84	157·60
½	15·809	31·618	47·427	63·236	79·045	94·854	110·66	126·47	142·28	158·09
⅝	15·857	31·714	47·571	63·428	79·285	95·142	111·00	126·86	142·71	158·57
¾	15·906	31·812	47·718	63·624	79·530	95·436	111·34	127·25	143·15	159·06
⅞	15·954	31·908	47·862	63·816	79·770	95·724	111·68	127·63	143·59	159·54
82	16·003	32·006	48·009	64·012	80·015	96·018	112·02	128·02	144·03	160·03
⅛	16·052	32·104	48·156	64·208	80·260	96·312	112·36	128·42	144·47	160·52
¼	16·101	32·202	48·303	64·404	80·505	96·606	112·71	128·81	144·91	161·01
⅜	16·150	32·300	48·450	64·600	80·750	96·900	113·05	129·20	145·35	161·50
½	16·199	32·398	48·597	64·796	80·995	97·194	113·39	129·59	145·79	161·99
⅝	16·248	32·496	48·744	64·992	81·240	97·488	113·74	129·98	146·23	162·48
¾	16·297	32·594	48·891	65·188	81·485	97·782	114·08	130·38	146·67	162·97
⅞	16·346	32·692	49·038	65·384	81·730	98·076	114·42	130·77	147·11	163·46
83	16·396	32·792	49·188	65·584	81·980	98·376	114·77	131·17	147·56	163·96
⅛	16·445	32·890	49·335	65·780	82·225	98·670	115·11	131·56	148·00	164·45
¼	16·495	32·990	49·485	65·980	82·475	98·970	115·46	131·96	148·45	164·95
⅜	16·544	33·088	49·632	66·176	82·720	99·264	115·81	132·35	148·90	165·44
½	16·594	33·188	49·782	66·376	82·970	99·564	116·16	132·75	149·35	165·94
⅝	16·644	33·288	49·932	66·576	83·220	99·864	116·51	133·15	149·80	166·44
¾	16·693	33·386	50·079	66·772	83·465	100·16	116·85	133·54	150·24	166·93
⅞	16·743	33·486	50·229	66·972	83·715	100·46	117·20	133·94	150·69	167·43
84	16·793	33·586	50·379	67·172	83·965	100·76	117·55	134·34	151·14	167·93
⅛	16·843	33·686	50·529	67·372	84·215	101·06	117·90	134·74	151·59	168·43
¼	16·893	33·786	50·679	67·572	84·465	101·36	118·25	135·14	152·04	168·93
⅜	16·944	33·888	50·832	67·776	84·720	101·66	118·61	135·55	152·50	169·44
½	16·994	33·988	50·982	67·976	84·970	101·96	118·96	135·95	152·95	169·94
⅝	17·044	34·088	51·132	68·176	85·220	102·26	119·31	136·35	153·40	170·44
¾	17·094	34·188	51·282	68·376	85·470	102·56	119·66	136·75	153·85	170·94
⅞	17·145	34·290	51·435	68·580	85·725	102·87	120·01	137·16	154·30	171·45

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
85	17·195	34·390	51·585	68·780	85·975	103·17	120·36	137·56	154·75	171·95
⅛	17·246	34·492	51·738	68·984	86·230	103·48	120·72	137·97	155·21	172·46
¼	17·297	34·594	51·891	69·188	86·485	103·78	121·08	138·38	155·67	172·97
⅜	17·348	34·696	52·044	69·392	86·740	104·09	121·44	138·78	156·13	173·48
½	17·398	34·796	52·194	69·592	86·990	104·39	121·79	139·18	156·58	173·98
⅝	17·449	34·898	52·347	69·796	87·245	104·69	122·14	139·59	157·04	174·49
¾	17·500	35·000	52·500	70·000	87·500	105·00	122·50	140·00	157·50	175·00
⅞	17·551	35·102	52·653	70·204	87·755	105·31	122·86	140·41	157·96	175·51
86	17·602	35·204	52·806	70·408	88·010	105·61	123·21	140·82	158·42	176·02
⅛	17·654	35·308	52·962	70·616	88·270	105·92	123·58	141·23	158·89	176·54
¼	17·705	35·410	53·115	70·820	88·525	106·23	123·93	141·64	159·34	177·05
⅜	17·756	35·512	53·268	71·024	88·780	106·54	124·29	142·05	159·80	177·56
½	17·808	35·616	53·424	71·232	89·040	106·85	124·66	142·46	160·27	178·08
⅝	17·859	35·718	53·577	71·436	89·295	107·15	125·01	142·87	160·73	178·59
¾	17·911	35·822	53·733	71·644	89·555	107·47	125·38	143·29	161·20	179·11
⅞	17·962	35·924	53·886	71·848	89·810	107·77	125·73	143·70	161·66	179·62
87	18·014	36·028	54·042	72·056	90·070	108·08	126·10	144·11	162·13	180·14
⅛	18·066	36·132	54·198	72·264	90·330	108·40	126·46	144·53	162·59	180·66
¼	18·118	36·236	54·354	72·472	90·590	108·71	126·83	144·94	163·06	181·18
⅜	18·170	36·340	54·510	72·680	90·850	109·02	127·19	145·36	163·53	181·70
½	18·222	36·444	54·666	72·888	91·110	109·33	127·55	145·78	164·00	182·22
⅝	18·274	36·548	54·822	73·096	91·370	109·64	127·92	146·19	164·47	182·74
¾	18·326	36·652	54·978	73·304	91·630	109·96	128·28	146·61	164·93	183·26
⅞	18·378	36·756	55·134	73·512	91·890	110·27	128·65	147·02	165·40	183·78
88	18·431	36·862	55·293	73·724	92·155	110·59	129·02	147·45	165·88	184·31
⅛	18·483	36·966	55·449	73·932	92·415	110·90	129·38	147·86	166·35	184·83
¼	18·536	37·072	55·608	74·144	92·680	111·22	129·75	148·29	166·82	185·36
⅜	18·588	37·176	55·764	74·352	92·940	111·53	130·12	148·70	167·29	185·88
½	18·641	37·282	55·923	74·564	93·205	111·85	130·49	149·13	167·77	186·41
⅝	18·693	37·386	56·079	74·772	93·465	112·16	130·85	149·54	168·24	186·93
¾	18·746	37·492	56·238	74·984	93·730	112·48	131·22	149·97	168·71	187·46
⅞	18·799	37·598	56·397	75·196	93·995	112·79	131·59	150·39	169·19	187·99

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
89	18·852	37·704	56·556	75·408	94·260	113·11	131·96	150·82	169·67	188·52
⅛	18·905	37·810	56·715	75·620	94·525	113·43	132·33	151·24	170·14	189·05
¼	18·958	37·916	56·874	75·832	94·790	113·75	132·71	151·66	170·62	189·58
⅜	19·011	38·022	57·033	76·044	95·055	114·07	133·08	152·09	171·10	190·11
½	19·064	38·128	57·192	76·256	95·320	114·38	133·45	152·51	171·58	190·64
⅝	19·118	38·236	57·354	76·472	95·590	114·71	133·83	152·94	172·06	191·18
¾	19·171	38·342	57·513	76·684	95·855	115·03	134·20	153·37	172·54	191·71
⅞	19·224	38·448	57·672	76·896	96·120	115·34	134·57	153·79	173·02	192·24
90	19·278	38·556	57·834	77·112	96·390	115·67	134·95	154·22	173·50	192·78
⅛	19·332	38·664	57·996	77·328	96·660	115·99	135·32	154·66	173·99	193·32
¼	19·385	38·770	58·155	77·540	96·925	116·31	135·69	155·08	174·46	193·85
⅜	19·439	38·878	58·317	77·756	97·195	116·63	136·07	155·51	174·95	194·39
½	19·493	38·986	58·479	77·972	97·465	116·96	136·45	155·94	175·44	194·93
⅝	19·547	39·094	58·641	78·188	97·735	117·28	136·83	156·38	175·92	195·47
¾	19·601	39·202	58·803	78·404	98·005	117·61	137·21	156·81	176·41	196·01
⅞	19·655	39·310	58·965	78·620	98·275	117·93	137·58	157·24	176·89	196·55
91	19·709	39·418	59·127	78·836	98·545	118·25	137·96	157·67	177·38	197·09
⅛	19·763	39·526	59·289	79·052	98·815	118·58	138·34	158·10	177·87	197·63
¼	19·817	39·634	59·451	79·268	99·085	118·90	138·72	158·54	178·35	198·17
⅜	19·872	39·744	59·616	79·488	99·360	119·23	139·10	158·98	178·85	198·72
½	19·926	39·852	59·778	79·704	99·630	119·56	139·48	159·41	179·33	199·26
⅝	19·980	39·960	59·940	79·920	99·900	119·88	139·86	159·84	179·82	199·80
¾	20·035	40·070	60·105	80·140	100·17	120·21	140·24	160·28	180·31	200·35
⅞	20·090	40·180	60·270	80·360	100·45	120·54	140·63	160·72	180·81	200·90
92	20·144	40·288	60·432	80·576	100·72	120·86	141·01	161·15	181·30	201·44
⅛	20·199	40·398	60·597	80·796	100·99	121·19	141·39	161·59	181·79	201·99
¼	20·254	40·508	60·762	81·016	101·27	121·52	141·78	162·03	182·29	202·54
⅜	20·309	40·618	60·927	81·236	101·54	121·85	142·16	162·47	182·78	203·09
½	20·364	40·728	61·092	81·456	101·82	122·18	142·55	162·91	183·28	203·64
⅝	20·419	40 838	61·257	81·676	102·09	122·51	142·93	163·35	183·77	204·19
¾	20·474	40·948	61·422	81·896	102·37	122·84	143·32	163·79	184·27	204·74
⅞	20·529	41·058	61·587	82·116	102·64	123·17	143·70	164·23	184·76	205·29

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
93	20·585	41·170	61·755	82·340	102·92	123·51	144·09	164·68	185·26	205·85
⅛	20·640	41·280	61·920	82·560	103·20	123·84	144·48	165·12	185·76	206·40
¼	20·695	41·390	62·085	82·780	103·47	124·17	144·86	165·56	186·25	206·95
⅜	20·751	41·502	62·253	83·004	103·75	124·51	145·26	166·01	186·76	207·51
½	20·807	41·614	62·421	83·228	104·03	124·84	145·65	166·46	187·26	208·07
⅝	20·862	41·724	62·586	83·448	104·31	125·17	146·03	166·90	187·76	208·62
¾	20·918	41·836	62·754	83·672	104·59	125·51	146·43	167·34	188·26	209·18
⅞	20·974	41·948	62·922	83·896	104·87	125·84	146·82	167·79	188·77	209·74
94	21·030	42·060	63·090	84·120	105·15	126·18	147·21	168·24	189·27	210·30
⅛	21·086	42·172	63·258	84·344	105·43	126·51	147·60	168·69	189·77	210·86
¼	21·142	42·284	63·426	84·568	105·71	126·85	147·99	169·14	190·28	211·42
⅜	21·198	42·396	63·594	84·792	105·99	127·19	148·39	169·58	190·78	211·98
½	21·254	42·508	63·762	85·016	106·27	127·52	148·78	170·03	191·29	212·54
⅝	21·310	42·620	63·930	85·240	106·55	127·86	149·17	170·48	191·79	213·10
¾	21·367	42·734	64·101	85·468	106·83	128·20	149·57	170·94	192·30	213·67
⅞	21·423	42·846	64·269	85·692	107·11	128·54	149·96	171·38	192·81	214·23
95	21·479	42·958	64·437	85·916	107·39	128·87	150·35	171·83	193·31	214·79
⅛	21·516	43·072	64·608	86·144	107·68	129·22	150·75	172·29	193·82	215·36
¼	21·593	43·186	64·779	86·372	107·96	129·56	151·15	172·74	194·34	215·93
⅜	21·649	43·298	64·947	86·596	108·24	129·89	151·54	173·19	194·84	216·49
½	21·706	43·412	65·118	86·824	108·53	130·24	151·94	173·65	195·35	217·06
⅝	21·763	43·526	65·289	87·052	108·81	130·58	152·34	174·10	195·87	217·63
¾	21·820	43·640	65·460	87·280	109·10	130·92	152·74	174·56	196·38	218·20
⅞	21·877	43·754	65·631	87.508	109·38	131·26	153·14	175·02	196·89	218·77
96	21·934	43·868	65·802	87·736	109·67	131·60	153·54	175·47	197·41	219·34
⅛	21·991	43·982	65·973	87·964	109·95	131·95	153·94	175·93	197·92	219·91
¼	22·048	44·096	66·144	88·192	110·24	132·29	154·34	176·38	198·43	220·48
⅜	22·106	44·212	66·318	88·424	110·53	132·64	154·74	176·85	198·95	221·06
½	22·163	44·326	66·489	88·652	110·81	132·98	155·14	177·30	199·47	221·63
⅝	22·221	44·442	66·663	88·884	111·10	133·33	155·55	177·77	199·99	222·21
¾	22·278	44·556	66·834	89·112	111·39	133·67	155·95	178·22	200·50	222·78
⅞	22·336	44·672	67·008	89·344	111·68	134·02	156·35	178·69	201·02	223·36

Indicated H.P. per lb. of Mean Pressure on Piston—*Continued.*

Diam. of cylndr ins.	Speed of Piston in Feet per Minute.									
	100	200	300	400	500	600	700	800	900	1000
97	22·393	44·786	67·179	89·572	111·96	134·36	156·75	179·14	201·54	223·93
1/8	22·451	44·902	67·353	89·804	112·25	134·71	157·16	179·61	202·06	224·51
1/4	22·509	45·018	67·527	90·036	112·54	135·05	157·56	180·07	202·58	225·09
3/8	22·567	45·134	67·701	90·268	112·83	135·40	157·97	180·54	203·10	225·67
1/2	22·625	45·250	67·875	90·500	113·12	135·75	158·37	181·00	203·62	226·25
5/8	22·683	45·366	68·049	90·732	113·41	136·10	158·78	181·46	204·15	226·83
3/4	22·741	45·482	68·223	90·964	113·70	136·45	159·19	181·93	204·67	227·41
7/8	22·799	45·598	68·397	91·196	113·99	136·79	159·59	182·39	205·19	227·99
98	22·857	45·714	68·571	91·428	114·28	137·14	160·00	182·86	205·71	228·57
1/8	22·916	45·832	68·748	91·664	114·58	137·50	160·41	183·33	206·24	229·16
1/4	22·974	45·948	68·922	91·896	114·87	137·84	160·82	183·79	206·77	229·74
3/8	23·033	46·066	69·099	92·132	115·16	138·20	161·23	184·26	207·30	230·33
1/2	23·091	46·182	69·273	92·364	115·45	138·55	161·64	184·73	207·82	230·91
5/8	23·150	46·300	69·450	92·600	115·75	138·90	162·05	185·20	208·35	231·50
3/4	23·209	46·418	69·627	92·836	116·04	139·25	162·46	185·67	208·88	232·09
7/8	23·267	46·534	69·801	93·068	116·33	139·60	162·87	186·14	209·40	232·67
99	23·326	46·652	69·978	93·304	116·63	139·96	163·28	186·61	209·93	233·26
1/8	23·385	46·770	70·155	93·540	116·92	140·31	163·69	187·08	210·46	233·85
1/4	23·444	46·888	70·332	93·776	117·22	140·66	164·11	187·55	211·00	234·44
3/8	23·503	47·006	70·509	94·012	117·51	141·02	164·52	188·02	211·53	235·03
1/2	23·563	47·126	70·689	94·252	117·81	141·38	164·94	188·50	212·07	235·63
5/8	23·622	47·244	70·866	94·488	118·11	141·73	165·35	188·98	212·59	236·22
3/4	23·681	47·362	71·043	94·724	118·40	142·09	165·77	189·45	213·13	236·81
7/8	23·740	47·480	71·220	94·960	118·70	142·44	166·18	189·92	213·66	237·40
100	23·800	47·600	71·400	95·200	119·00	142·80	116·60	190·40	214·20	238·00

THE PROPERTIES OF SATURATED STEAM.

Temperature of the steam and water.	Number of Thermal Units contained in 1 lb. Latent heat.	Number of Thermal Units contained in 1 lb. Total contained in the steam.	Pressure above zero expressed in pounds per square inch.	Vacuum by gauge in inches of mercury at 32 deg.	Weight of 1 cubic foot of steam.	Volume of 1 lb. of steam.
Deg.					Pounds.	Cubic feet.
60	1072·2	1132·2	·26	29·40	·0008	1220
61	1071·5	1132·5	·26	29·38	·0008	1176
62	1070·8	1132·8	·27	29·36	·0009	1136
63	1070·1	1133·2	·28	29·34	·0009	1099
64	1069·4	1133·5	·29	29·32	·0009	1064
65	1068·8	1133·8	·30	29·30	·0010	1031
66	1068·1	1134·1	·31	29·28	·0010	1000
67	1067·4	1134·4	·32	29·26	·0010	970·8
68	1066·7	1134·7	·33	29·23	·0011	934·6
69	1065·9	1135·0	·35	29·21	·0011	901·0
70	1065·3	1135·3	·36	29·19	·0011	869·5
71	1064·6	1135·6	·37	29·16	·0012	840·4
72	1063·9	1135·9	·38	29·14	·0012	813·0
73	1063·2	1136·2	·40	29·11	·0013	787·4
74	1062·5	1136·5	·41	29·08	·0013	763·4
75	1061·8	1136·8	·42	29·05	·0013	740·7
76	1061·1	1137·1	·44	29·02	·0014	719·4
77	1060·4	1137·4	·45	28·99	·0014	699·3
78	1059·7	1137·7	·47	28·96	·0015	675·6
79	1059·0	1138·0	·49	28·93	·0015	653·6
80	1058·3	1138·3	·50	28·90	·0016	632·9
81	1057·6	1138·6	·52	28·86	·0016	613·5
82	1056·9	1138·9	·53	28·83	·0017	595·2
83	1056·2	1139·3	·55	28·79	·0017	578·1
84	1055·5	1139·6	·57	28·76	·0018	561·8
85	1054·8	1139·9	·59	28·72	·0018	546·5
86	1054·1	1140·2	·61	28·68	·0019	529·1
87	1053·4	1140·5	·63	28·64	·0019	512·8
88	1052·7	1140·8	·65	28·60	·0020	497·6
89	1052·0	1141·1	·67	28·55	·0021	483·1
90	1051·3	1141·4	·69	28·51	·0021	469·5
91	1050·6	1141·7	·71	28·47	·0022	456·6
92	1049·9	1142·0	·74	28·42	·0023	442·5
93	1049·2	1142·3	·76	28·37	·0023	429·2
94	1048·5	1142·6	·78	28·32	·0024	416·7
95	1047·8	1142·9	·81	28·27	·0025	404·8
96	1047·2	1143·2	·83	28·22	·0025	393·7
97	1046·5	1143·5	·86	28·17	·0026	381·7
98	1045·8	1143·8	·89	28·12	·0027	370·4
99	1045·1	1144·1	·91	28·06	·0028	358·8

The Properties of Saturated Steam—*continued.*

Temperature of the steam and water.	Number of Thermal Units contained in 1 lb. — Latent heat.	Number of Thermal Units contained in 1 lb. — Total contained in the steam.	Pressure above zero expressed in pounds per square inch.	Vacuum by gauge in inches of mercury at 32 deg.	Weight of 1 cubic foot of steam.	Volume of 1 lb. of steam.
Deg.					Pounds.	Cubic feet.
100	1044·4	1144·4	·94	28·00	·0029	349·6
101	1043·7	1144·7	·97	27·94	·0029	340·1
102	1043·0	1145·0	1·00	27·88	·0030	331·1
103	1042·3	1145·4	1·03	27·82	·0031	321·6
104	1041·6	1145·7	1·06	27·76	·0032	312·5
105	1040·9	1146·0	1·09	27·69	·0033	303·0
106	1040·2	1146·3	1·13	27·63	·0034	294·1
107	1039·5	1146·6	1·15	27·56	·0035	285·7
108	1038·8	1146·9	1·19	27·49	·0036	277·8
109	1038·1	1147·2	1·23	27·42	·0037	270·3
110	1037·4	1147·5	1·26	27·34	·0038	263·2
111	1036·7	1147·8	1·30	27·27	·0039	256·4
112	1036·0	1148·1	1·34	27·19	·0040	250·0
113	1035·3	1148·4	1·38	27·11	·0041	243·9
114	1034·6	1148·7	1·42	27·03	·0042	237·5
115	1033·9	1149·0	1·46	26·94	·0043	231·0
116	1033·2	1149·3	1·50	26·86	·0044	224·7
117	1032·5	1149·6	1·55	26·77	·0046	218·8
118	1031·8	1149·9	1·59	26·68	·0047	212·8
119	1031·1	1150·2	1·64	26·59	·0048	207·0
120	1030·4	1150·5	1·68	26·50	·0050	201·6
121	1029·7	1150·9	1·73	26·40	·0051	196·9
122	1029·0	1151·1	1·78	26·30	·0052	192·0
123	1028·2	1151·5	1·83	26·20	·0053	186·9
124	1027·6	1151·8	1·88	26·09	·0055	182·7
125	1026·9	1152·1	1·93	26·00	·0056	177·6
126	1026·2	1152·4	1·98	25·88	·0058	173·0
127	1025·5	1152·7	2·04	25·77	·0059	168·6
128	1024·8	1153·0	2·10	25·65	·0061	164·5
129	1024·1	1153·3	2·15	25·54	·0062	160·3
130	1023·4	1153·6	2·21	25·42	·0064	156·3
131	1022·7	1153·9	2·27	25·30	·0066	152·5
132	1022·0	1154·2	2·33	25·17	·0067	148·6
133	1021·3	1154·5	2·40	25·04	·0069	144·9
134	1020·6	1154·8	2·46	24·91	·0071	141·4
135	1019·9	1155·1	2·53	24·78	·0072	137·9
136	1019·2	1155·4	2·59	24·64	·0074	134·6
137	1018·5	1155·7	2·66	24·50	·0076	131·5
138	1017·8	1156·0	2·73	24·36	·0078	128·2
139	1017·1	1156·3	2·80	24·21	·0080	125·2

The Properties of Saturated Steam—*continued.*

Temperature of the steam and water.	Number of Thermal Units contained in 1 lb. — Latent heat.	Number of Thermal Units contained in 1 lb. — Total contained in the steam.	Pressure above zero expressed in pounds per square inch.	Vacuum by gauge in inches of mercury at 32 deg.	Weight of 1 cubic foot of steam.	Volume of 1 lb. of steam.
Deg.					Pounds.	Cubic feet.
140	1016·4	1156·6	2·88	24·06	·0082	122·1
141	1015·7	1156·9	2·95	23·91	·0084	119·2
142	1015·0	1157·2	3·03	23·75	·0086	116·3
143	1014 3	1157·6	3·11	23·59	·0088	113·5
144	1013·6	1157·9	3·19	23·43	·0090	110·7
145	1012·9	1158·2	3·27	23·26	·0092	108·1
146	1012·2	1158·5	3·35	23·09	·0095	105·5
147	1011·5	1158·8	3·44	22·92	·0097	103·0
148	1010·8	1159·1	3·53	22·74	·0099	100·7
149	1010·1	1159·4	3·61	22·56	·0102	98·42
150	1009·4	1159·7	3·71	22·37	·0104	96·16
151	1008·7	1160·0	3·80	22·18	·0106	93·99
152	1008·0	1160·3	3·90	21·99	·0109	91·84
153	1007·3	1160·6	3·99	21·79	·0111	89·78
154	1006·6	1160·9	4·09	21·59	·0114	87·72
155	1005·9	1161·2	4·19	21·39	·0117	85·70
156	1005·2	1161·5	4·29	21·18	·0119	83·76
157	1004·5	1161·8	4·40	20·96	·0122	81·84
158	1003·8	1162·1	4·51	20·74	·0125	80·00
159	1003·1	1162·4	4·62	20·52	·0128	78 19
160	1002·4	1162·7	4·73	20·29	·0131	76·46
161	1001·7	1163·0	4·84	20·06	·0134	74·75
162	1001·0	1163·3	4·96	19·82	·0137	73·10
163	1000·3	1163·7	5·08	19·58	·0140	71·48
164	999·6	1164·0	5·20	19·33	·0143	69·94
165	998·9	1164·3	5·32	19·08	·0146	68·41
166	998·1	1164·6	5·45	18·82	·0149	66·90
167	997·4	1164·9	5·58	18·56	·0153	65·46
168	996·7	1165·2	5·71	18·29	·0156	64·02
169	996·0	1165·5	5·84	18·02	·0160	62·66
170	995·3	1165·8	5·98	17·74	·0163	61·31
171	994·6	1166·1	6·12	17·46	·0167	59·99
172	993·9	1166·4	6·26	17·17	·0170	58·69
173	993·2	1166·7	6·41	16·87	·0174	57·45
174	992·5	1167·0	6·55	16·58	·0178	56·21
175	991·8	1167·3	6·70	16·27	·0182	55·19
176	991·1	1167·6	6·86	15·96	·0185	53·91
177	990·4	1167·9	7·01	15·64	·0189	52·80
178	989·7	1168·2	7·17	15·32	·0193	51·71
179	989·0	1168·5	7·33	14·99	·0197	50·64
180	988·3	1168·9	7·50	14·65	·0202	49·6

TABLE OF PROPERTIES OF SATURATED STEAM.

Absolute pressure in lbs. per square inch.	Temperature or boiling point in degrees Fah.	Total heat contained in 1lb. steam, including heating liquid water from 0° F. British thermal units.	Weight of 1 cub. ft. of steam in lbs.	Volume of 1lb. weight of steam in cub. ft.	Cubic feet of steam from 1 cub. ft. of water at 62° F.
1	102·1	1145·0	·0030	330·36	20,600
2	126·3	1152·2	·0058	172·08	10,730
3	141·6	1156·8	·0085	117·52	7,327
4	153·1	1160·1	·0112	89·62	5,589
5	162·3	1163·0	·0138	72·66	4,530
6	170·2	1165·3	·0163	61·21	3,816
7	176·9	1167·3	·0189	52·94	3,301
8	182·9	1169·2	·0214	46·69	2,911
9	188·3	1170·8	·0239	41·79	2,606
10	193·3	1172·3	·0264	37·84	2,360
11	197·8	1173·7	·0289	34·62	2,157
12	202·0	1175·0	·0314	31·88	1,988
13	205·9	1176·2	·0338	29·27	1,844
14	209·6	1177·3	·0362	27·61	1,721
14·7	212·0	......	·0380	26·36	1,644
15	213·1	1178·4	·0387	25·85	1,611
20	228·0	1182·9	·0507	19·72	1,229
25	240·1	1186·6	·0625	15·99	996
30	250·4	1189·8	·0743	13·46	838
35	259·3	1192·5	·0858	11·65	726
40	267·3	1194·9	·0974	10·27	640
45	274·4	1197·1	·1089	9·18	572
50	281	1199·1	·1202	8·31	518
55	287·1	1201·0	·1314	7·61	474
60	292·7	1202·7	·1425	7·01	437
65	298	1204·3	·1538	6·49	405

PROPERTIES OF SATURATED STEAM—*continued.*

Absolute pressure in lbs. per square inch.	Temperature or boiling point in degrees Fah.	Total heat contained in 1lb. steam, including heating liquid water from 0°F. British thermal units.	Weight of 1 cub. ft. of steam in lbs.	Volume of 1lb. weight of steam in cub. ft.	Cubic feet of steam from 1 cub. ft. of water at 62°F.
70	302·9	1205·8	·1648	6·07	378
75	307·5	1207·2	·1759	5·68	353
80	312·0	1208·5	·1869	5·35	333
85	316·1	1209·9	·1980	5·05	314
90	320·0	1211·1	·2089	4·79	298
95	324·1	1212·3	·2198	4·55	283
100	327·9	1213·4	·2307	4·33	270
105	331·3	1214·4	·2414	4·14	257
110	334·6	1215·5	·2521	3·97	247
115	338·0	1216·5	·2628	3·80	237
120	341·1	1217·4	·2738	3·65	227
125	344·2	1218·4	·2845	3·51	219
130	347·2	1219·3	·2955	3·38	211
135	350·1	1220·2	·3060	3·27	203
140	352·9	1221·0	·3162	3·16	197
145	355·6	1221·9	·3273	3·06	190
150	358·3	1222·7	·3377	2·96	184
155	361·0	1223·5	·3484	2·87	179
160	364·3	1224·2	·3590	2·79	174
165	366·0	1224·9	·3695	2·71	169
170	368·2	1225·7	·3798	2·63	164
175	370·8	1226·4	·3899	2·56	159
180	372·9	1227·1	·4009	2·49	155
185	375·3	1227·8	·4117	2·43	151
190	377·5	1228·5	·4222	2·37	148
195	379·7	1229·2	·4327	2·31	144
200	381·7	1229·8	·4431	2·26	141

AREAS OF CIRCLES, ADVANCING BY EIGHTHS.

Dia. In.	AREAS.								Dia. In.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
0	·0	·012	·049	·11	·196	·306	·441	·601	0
1	·785	·994	1·227	1·484	1·767	2·073	2·405	2·761	1
2	3·141	3·546	3·976	4·43	4·908	5·411	5·939	6·491	2
3	7·068	7·669	8·295	8·946	9·621	10·32	11·04	11·79	3
4	12·56	13·36	14·18	15·03	15·9	16·8	17·72	18·66	4
5	19·63	20·62	21·64	22·69	23·75	24·85	25·96	27·1	5
6	28·27	29·46	30·67	31·91	33·18	34·47	35·78	37·12	6
7	38·48	39·87	41·28	42·71	44·17	45·66	47·17	48·7	7
8	50·26	51·84	53·45	55·08	56·74	58·42	60·13	61·86	8
9	63·61	55·39	67·2	69·02	70·88	72·75	74·66	76·58	9
10	78·54	80·51	82·51	84·54	86·59	88·66	90·76	92·88	10
11	95·03	97·2	99·5	101·62	103·86	106·13	108·43	110·75	11
12	113·09	115·46	117·85	120·27	122·71	125·18	127·67	130·19	12
13	132·73	135·29	137·88	140·5	143·13	145·8	148·49	151·2	13
14	153·93	156·7	159·48	162·29	165·13	167·99	170·87	173·78	14
15	176·71	179·67	182·65	185·66	188·69	191·74	194·82	197·93	15
16	201·06	204·21	207·39	210·59	213·82	217·07	220·35	223·65	16
17	226·98	230·33	233·70	237·1	240·52	243·97	247·45	250·94	17
18	254·47	258·01	261·58	265·18	268·8	272·44	276·11	279·81	18
19	283·52	287·27	291·04	294·83	298·64	302·48	306·35	310·24	19
20	314·16	318·09	322·06	326·05	330·06	334·1	338·16	342·25	20
21	346·36	350·49	354·65	358·84	363·05	367·28	371·54	375·82	21
22	380·13	384·46	388·82	393·2	397·6	402·03	406·49	410·97	22
23	415·47	420	424·55	429·13	433·73	438·36	443·01	447·69	23
24	452·39	457·11	461·86	466·63	471·43	476·25	481·1	485·97	24
25	490·87	495·79	500·74	505·71	510·7	515·72	520·76	525·83	25
26	530·93	536·04	541·19	546·35	551·54	556·76	562	567·26	26
27	572·55	577·87	583·2	588·57	593·95	599·37	604·8	610·26	27
28	615·75	621·26	626·79	632·35	637·94	643·54	649·18	654·84	28
29	660·52	666·22	671·95	677·71	683·49	689·29	695·12	700·98	29
30	706·86	712·76	718·69	724·64	730·61	736·61	742·64	748·69	30
31	754·79	760·87	766·99	773·14	779·31	785·51	791·73	797·97	31
32	804·25	810·54	816·86	823·21	829·57	835·97	842·39	848·83	32
33	855·3	861·79	868·3	874·85	881·41	888	894·62	901·25	33
34	907·92	914·61	921·32	928·06	934·82	941·6	948·42	955·25	34
35	962·11	969	975·9	982·84	989·8	996·78	1003·79	1010·82	35
36	1017·87	1024·96	1032·06	1039·19	1046·34	1053·52	1060 72	1067·96	36
37	1075·21	1082·49	1089·79	1097·11	1104·46	1111·84	1119·24	1126·66	37
38	1134·11	1141·59	1149·08	1156·61	1164·15	1171·73	1179·32	1186·94	38
39	1194·59	1202·26	1209·95	1217·67	1225·42	1233·18	1240·98	1248·79	39
40	1256·64	1264·51	1272·4	1280·31	1288·25	1296·22	1304·21	1312·22	40
41	1320·26	1328·32	1336·41	1344·52	1352·66	1360·82	1369	1377·21	41
42	1385·45	1393·7	1401·99	1410·3	1418·63	1426·99	1435·37	1443·77	42
43	1452·2	1460 66	1469·14	1477·64	1486·17	1494·73	1503·3	1511·91	43
44	1520·53	1529·19	1537·86	1546·56	1555·29	1564·04	1572·81	1581·61	44
45	1590·43	1599·28	1608·16	1617·05	1625·97	1634·92	1643·89	1652·89	45
46	1661·91	1670·95	1680·02	1689·11	1698·23	1707·37	1716·54	1725·73	46
47	1734·95	1744·19	1753·45	1762·74	1772·06	1781·4	1790·76	1800·15	47
48	1809·56	1819	1828·46	1837·95	1847·46	1856·99	1866·55	1876·14	48
49	1885·75	1895·38	1905·04	1914·72	1924·43	1934·16	1943·91	1953·69	49

AREAS OF CIRCLES.—*Continued.*

Dia. In.	AREAS.								Dia. In.
	·0	·⅛	·¼	·⅜	·½	·⅝	·¾	·⅞	
50	1963·5	1973·3	1983·1	1993	2002·9	2012·8	2022·8	2032·8	50
51	2042·8	2052·8	2062·9	2072·9	2083	2093·2	2103·5	2113·5	51
52	2123·7	2133·9	2144·1	2154·4	2164·7	2175	2185·4	2195·7	52
53	2206·1	2216·6	2227	2237·5	2248	2258·5	2269	2279·6	53
54	2290·2	2300·8	2311·4	2322·1	2332·8	2343·5	2354·2	2365	54
55	2375·8	2386·6	2397·4	2408·3	2419·2	2430·1	2441	2452	55
56	2463	2474	2485	2496·1	2507·1	2518·3	2529·4	2540·5	56
57	2551·7	2562·9	2574	2585·4	2596·7	2608	2619·3	2630·7	57
58	2642	2653·4	2664·9	2676·3	2687·8	2699·3	2710·8	2722·4	58
59	2733·9	2745·5	2757·2	2768·8	2780·5	2792·2	2803·9	2815·6	59
60	2827·4	2839·2	2851	2862·8	2874·7	2886·6	2898·5	2910·5	60
61	2922·4	2934·4	2946·4	2958·5	2970·5	2982·6	2994·7	3006·9	61
62	3019	3031·2	3043·4	3055·7	3067·9	3080·2	3092·5	3104·8	62
63	3117·2	3129·6	3142	3154·4	3166·9	3179·4	3191·9	3204·4	63
64	3217	3229·5	3242·1	3254·8	3267·4	3280·1	3292·8	3305·5	64
65	3318·3	3331	3343·8	3356·7	3369·5	3382·4	3395·3	3408·2	65
66	3421·2	3434·1	3447·1	3460·1	3473·2	3486·3	3499·4	3512·5	66
67	3525·6	3538·8	3552	3565·2	3578·4	3591·7	3605	3618·3	67
68	3631·6	3645	3658·4	3671·8	3685·2	3698·7	3712·2	3725·7	68
69	3739·2	3752·8	3766·4	3780	3793·6	3807·3	3821	3834·7	69
70	3848·4	3862·2	3876	3889·8	3903·6	3917·4	3931·3	3945·2	70
71	3959·2	3973·1	3987·1	4001·1	4015·1	4029·2	4043·2	4057·3	71
72	4071·5	4085·6	4099·8	4114	4128·2	4142·5	4156·7	4171	72
73	4185·4	4199·7	4214·1	4228·5	4242·9	4257·3	4271·8	4286·3	73
74	4300·8	4315·3	4329·9	4344·5	4359·1	4373·8	4388·4	4403·1	74
75	4417·8	4432·6	4447·3	4462·1	4476·9	4491·8	4506·6	4521·5	75
76	4536·4	4551·4	4566·3	4581·3	4596·3	4611·3	4626·4	4641·5	76
77	4656·6	4671·7	4686·9	4702 1	4417·3	4732·5	4747·7	4763	77
78	4778·3	4793·7	4809	4824·4	4839·8	4855·2	4870·7	4886·1	78
79	4901·6	4917·2	4932·7	4948·3	4963·9	4979·5	4995·1	5010·8	79
80	5026·5	5042·2	5058	5073·7	5089·5	5105·4	5121·2	5137·1	80
81	5153	5168·9	5184·8	5200·8	5216·8	5232·8	5248·8	5264·9	81
82	5281	5297·1	5313·2	5329·4	5345·6	5361·8	5378	5394·3	82
83	5410·6	5426·9	5443·2	5459·6	5476	5492·4	5508·8	5525·3	83
84	5541·7	5558·2	5574·8	5591·3	5607·9	5624·5	5641·1	5657·8	84
85	5674·5	5691·2	5707·9	5724·6	5741·4	5758·2	5775·1	5791·9	85
86	5808·8	5825·7	5842·6	5859·5	5876·5	5893·5	5910·5	5927·6	86
87	5944·6	5961·7	5978·9	5996	6013·2	6030·4	6047·6	6064·8	87
88	6082·1	6099 4	6116·7	6134	6151·4	6166·8	6186·2	6203·6	88
89	6221·1	6238·6	6256·1	6273·6	6291·2	6308·8	6326·4	6344	89
90	6361·7	6379·4	6397·1	6414·8	6432·6	6450·4	6468·2	6486	90
91	6503·9	6521·7	6539·6	6557·6	6575·5	6593·5	6611·5	6629·5	91
92	6647·6	6665·7	6683·8	6701·9	6720	6738·2	6756·4	6774·6	92
93	6792·9	6811·2	6829·4	6847·8	6866·1	6884·5	6902·9	6921·3	93
94	6939·7	6958·2	6976·7	6995·2	7013·8	7032·3	7050·9	7069·5	94
95	7088·2	7106·9	7125·5	7144·3	7163	7181·8	7200·6	7219·4	95
96	7238·2	7257·1	7275·9	7294·9	7313·8	7332·8	7351·7	7370·7	96
97	7389·8	7408·8	7427·9	7447	7466·2	7485·3	7504·5	7523·7	97
98	7542·9	7562·2	7581·5	7600·8	7620·1	7639·5	7658·8	7678·2	98
99	7697·7	7717·1	7736·6	7756·1	7775·6	7795·2	7814·7	7834·3	99

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